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THE ENGLISH ELECTRIC JOURNAL

VOLUME TWELVE • NUMBER SIX • MARCH 1952

THE ENGLISH ELECTRIC JOURNAL

Their Majesties King George VI and the Queen arriving at Stafford Works on the 26th of February, 1942.

Editorial

As this issue of the Journal goes to press, the tragic news of the sudden passing of our beloved King George VI has been received by a sorrowing world.

Little can be added here to the just tributes which have already been paid to him by our Prime Minister, Mr. Winston Churchill, and other eminent men throughout the British Commonwealth of Nations, the Empire, and the World, but it is appropriate for us to recall His late Majesty's visits to the Company's Stafford and Liverpool Works during the war.

His personal identification with his peoples' efforts in those anxious days inspired the will to the final victory which happily he lived to see realised.

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CONTENTS

Page 3: NEW RAIL TRACTION ENGINES TYPE 'RK', *by J. M. Smith, B.Sc., A.M.I.Mech.E., M.I.Mar.E., Chief Engineer, Diesel Engine Division.*

14: GOVERNORS AND CONTROL GEAR FOR DIESEL ENGINES, *by L. Berman, M.A., A.M.I.Mech.E., Assistant Chief Engineer (Preston), Diesel Engine Division.*

21: THE MANUFACTURE OF ASBESTOS-CEMENT MATERIALS.

27: DESIGN AND OPERATION OF EXHAUST-DRIVEN TURBO-COMPRESSORS FOR DIESEL ENGINES, *by A. J. Penn, M.I.Mech.E., Chief Engineer, Aero Gas Turbine Division, D. Napier & Son Limited.*

37: THE NAPIER 'NOMAD' AERO ENGINE.

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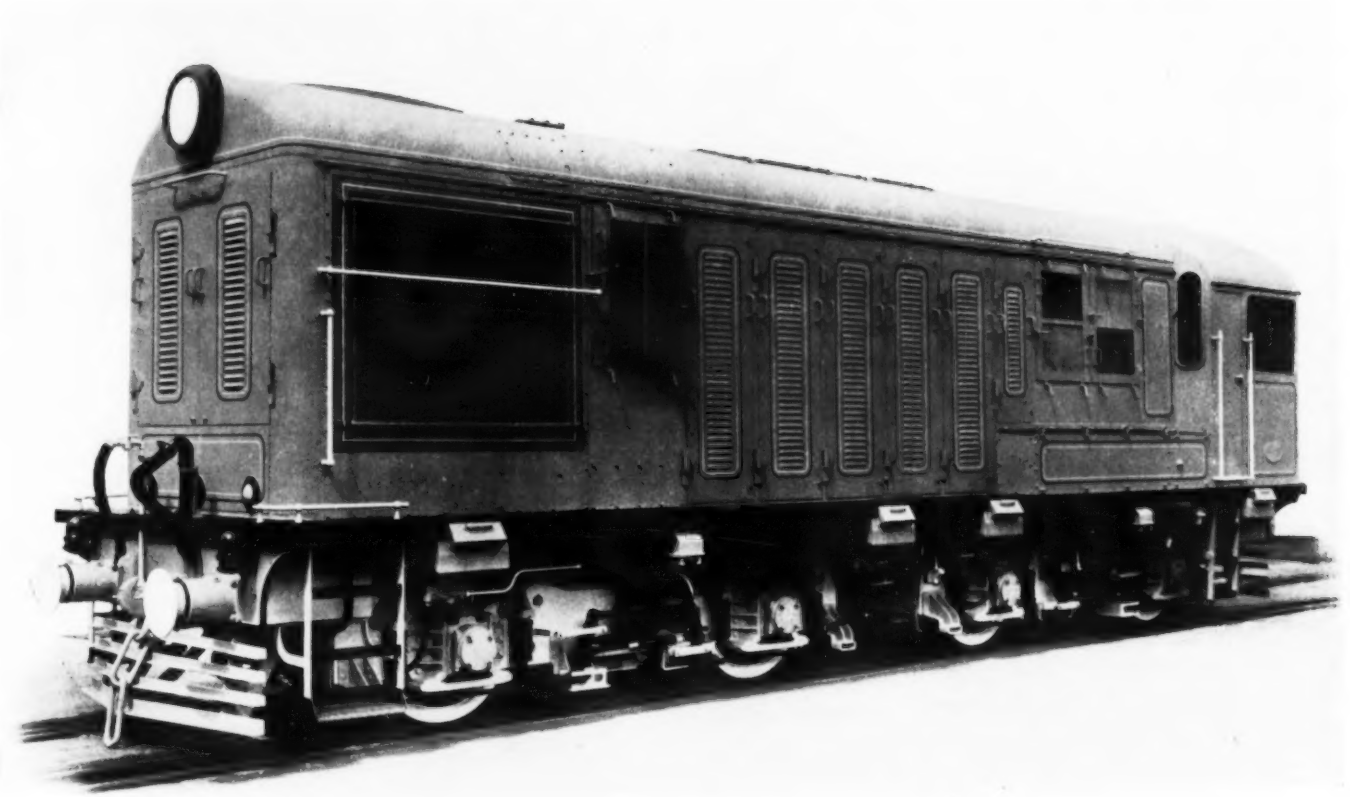


A 660 h.p. diesel-electric general service locomotive on the Tasmanian Government Railways

New Rail Traction Engines Type 'RK'

By J. M. SMITH, B.Sc., A.M.I.Mech.E., M.I.Mar.E.,
Chief Engineer, Diesel Engine Division.

IN ORDER TO describe the new 'RK' type rail traction diesel engines which are now in service, it is first necessary to refer to the 'V' type engines from which they were derived, and also to the 'K' type engines which preceded the 'V' type and had proved very successful in rail traction service.

THE 'V' AND 'K' ENGINES

The English Electric Company's 'V' type rail traction diesel engine was introduced a few years ago and achieved conspicuous success in the first main line diesel-electric locomotives built for use on British Railways. Two locomotives, No. 10,000 and No. 10,001, were put in hand by the London Midland and Scottish Railway and the first went into regular operation just before the nationalisation of the railways. Working together and separately, these locomotives have ably demonstrated the potentialities of diesel-electric traction in Britain ; they have established a great reputation and won high regard in railway circles and with the travelling public. They are now owned by the London Midland Region of British Railways, and others are being commissioned by the Southern Region of which the first, No. 10,201, was on view at the Festival of Britain South Bank Exhibition in London during 1951.

In these locomotives, and in a fleet of rather similar main line locomotives in service in Egypt, the motive power unit is the 16-cylinder 'V' engine. This Company was the first in Britain to develop a large medium-speed engine of this type*, and the engine has proved capable of the highest power outputs in relation to cylinder size, with great durability and impressive fuel economy.

* *As there is much of interest in the history of this development, it is intended to publish a separate article on that subject in a subsequent issue of this journal.*

The 'V' engine is made in 12-cylinder and 16-cylinder models having the same cylinder dimensions, i.e., 10 in. bore by 12 in. stroke, and with the cylinders arranged in two banks at an angle of 45 degrees. The basic cylinder size is exactly the same as in the earlier 'K' type engine, this size of cylinder being estimated to result in the most suitable overall engine dimensions and operating characteristics for powerful traction applications throughout the world's railway systems. The adoption of this cylinder size also enabled much of the experience and constructional technique obtained with the 'K' engine to be applied with certainty in the 'V' design, but full consideration showed that the latter would have been seriously handicapped if attempts had been made to embody in it many of the actual 'K' engine components. It was therefore decided that the 'V' engine should be designed throughout solely in terms of the Vee arrangement (just as the 'K' engine had been designed earlier, with only the in-line formation of cylinders in mind) and experience has shown the wisdom of this course.

With the 'V' type in being, the Company's range of traction engines with the same cylinder size comprised the 12- and 16-cylinder 'V' models, with structures and components designed for heavy mechanical and thermal loading under supercharging conditions and high-speed operation, and the 'K' type in-line engines with 6 and 8 cylinders which, although having the same cylinder size as the 'V' type, did not have the same cylinder components or the same potentialities for power development. As the application of supercharging was becoming more usual and was certain to be applied to increasing extent and degree, and it was most advantageous to be able to offer railway operators a wide range of engines employing the same component parts, the obvious course was to develop in-line versions of the 'V' design.

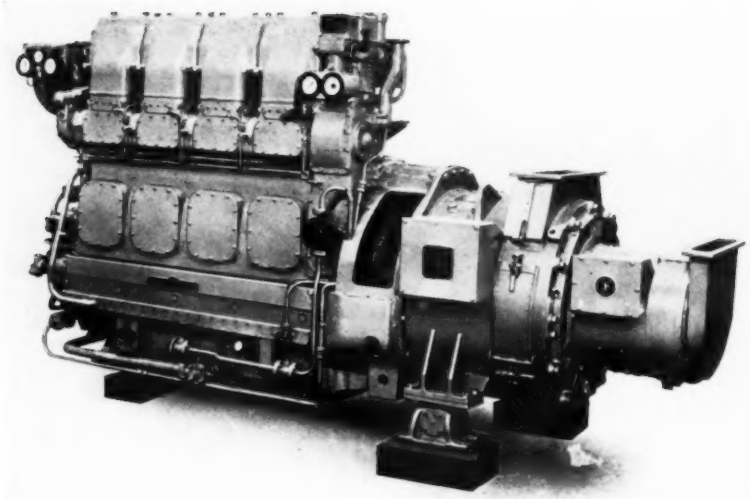


Fig. 1. A '4SRKT' pressure charged diesel engine and generator

THE NEW 'RK' ENGINES

In accordance with the foregoing considerations, the important step was taken of introducing 4-, 6- and 8-cylinder engines, designated 'RK,'† in which as many components as possible are common to the 'V' type, for although, as already stated, it would have been detrimental to impose existing in-line engine components on the 'V' design, there was no such disadvantage, in fact there was every advantage, in reversing the process. Components such as cylinder heads, valves and valve gear, cylinders and pistons, connecting rods and bearings, cams and camshaft drives, fuel injection equipment, etc., are common to all engines of both the 'V' and the new 'RK' types. The original 'V' design, with minor improvements, responded to all development demands, enabling the accumulated experience with the 'V' engines in research, development and service to be embodied in the new 'RK' engines to a degree which would not be possible in a completely new design.

The Company is therefore now in the unique position of being able to supply engines of 4, 6, 8, 12 or 16 cylinders with identical maintenance requirements and operating characteristics, representing the most fully integrated range of diesel traction engines ever produced and enabling operators to standardise on one basic engine type for all services from the shunting locomotive to the 100 m.p.h. main line passenger express locomotive.

Balancing

The three 'RK' traction engines (Figs. 1, 2 and 3)‡ are all of the vertical in-line four-stroke cycle type and, an important consideration for traction application, each has complete internal balance so that running is practically vibrationless and installation can be made with confidence of smooth operation on any type of rail vehicle, including such light structures as railcars. To secure complete balance of six- and eight-cylinder engines, only a strong engine frame and a certain disposition of the crankshaft throws are required in each case, but for four-cylinder engines complete balance is not so easily obtained and therefore large medium-speed engines of this kind have not hitherto been applicable to traction service because of the magnitude of the unbalanced forces.

The '4RK' engine is therefore an innovation for traction work. Its balancing principle, however, is orthodox, on the system devised by Lanchester.

† *Normally the new engines would have been designated 4V, 6V and 8V, indicating a specific design of cylinder rather than the Vee formation of cylinders, but as there were objections to using the symbol 'V' in this case, the letters 'RK' (re-designed K) were chosen, since the new engines would ultimately replace the original 'K' type having the same bore and stroke.*

‡ *The additional symbols 'S' and 'T' in the engine designations given in the captions signify 'supercharged' and 'traction' respectively, and the prefixed numeral indicates the number of cylinders.*

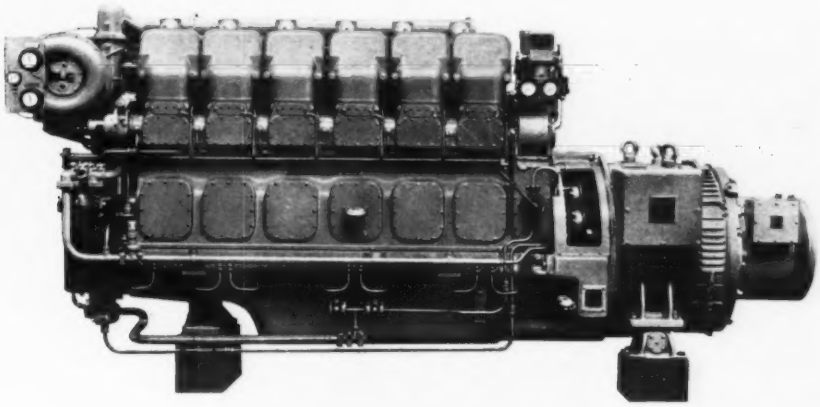


Fig. 2.
A '6SRKT' pressure charged diesel engine and generator

This depends on an arrangement of crankshaft throws which results in complete internal balance of primary forces and couples and of secondary couples, but leaves fully unbalanced the secondary vertical forces. These secondary forces of the crankshaft are cancelled by a system of two contra-rotating shafts with appropriate balance masses, incorporated in the engine structure and revolved at twice crankshaft speed. The engine is therefore as smooth in operation as the six- and eight-cylinder sizes and creates a most favourable impression on observers.

General Construction

Apart from this distinguishing feature, the balancing system, the '4RK' is similar in specification to the '6RK' and the '8RK,' and Figs. 4 and 5 show sectional views of an 8-cylinder supercharged 'RK' type engine. In each case the main structure of the engine is composed of

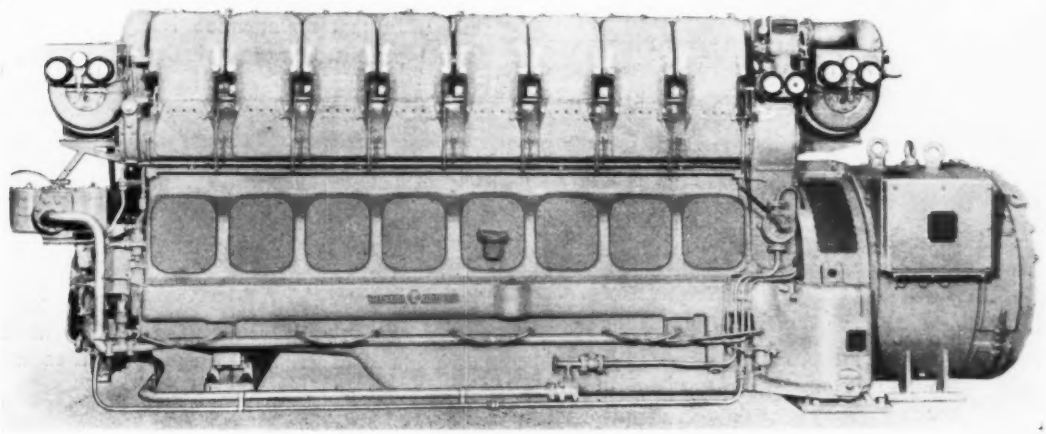


Fig. 3. An '8SRKT' pressure charged diesel engine and generator

two high-grade iron castings, the bedplate which contains the crankshaft bearing housings and the crankcase which comprises the upper structure of the crank chamber and a complete cylinder block. In the cylinder block separate detachable cylinder liners are fitted and surmounted by cylinder heads, also of cast iron construction.

At the non-driving end of the engine is fitted a strong cast gearbox for water and oil pump drives, and at the driving end the bedplate casting extends in a semi-cylindrical structure to which the traction generator is bolted. Additional support for the generator is afforded at the top by a strong strut between it and the crankcase end wall which is strengthened especially for this attachment. The complete diesel-electric power plant is therefore a single rigid unit of great strength which requires no additional provision for preserving shaft alignment, and is in fact normally supplied ready for installation on three-point supports to isolate it from bending or twisting moments consequent on movements of the framework of the vehicle in which it is mounted.

Moving Parts

The crankshaft and connecting rods are all high-grade alloy steel forgings fully heat treated and suitably machined, and the pistons are of aluminium Y-alloy. The gudgeon pins are of case hardened steel. They are carried directly on the metal of the pistons, in bores in which they are free only when the piston is hot, and are restrained from endwise movement by circlips registering in bore grooves. A phosphor bronze bush pressed into the eye of the connecting rod forms the small-end bearing in which the rod oscillates on the gudgeon pin. At the crankpin end the rod is split and the bore fitted with half bearings of steel lined with bearing metal. The cap half of the bearing is attached by four high-tensile bolts with nuts of special design to avoid stress concentration on the screw thread at the interface.

Bearings of similar construction are used to support the crankshaft in the bedplate, and the bolts which retain the bearing caps are also of alloy steel with the same type of special nut as the big-end bolts.

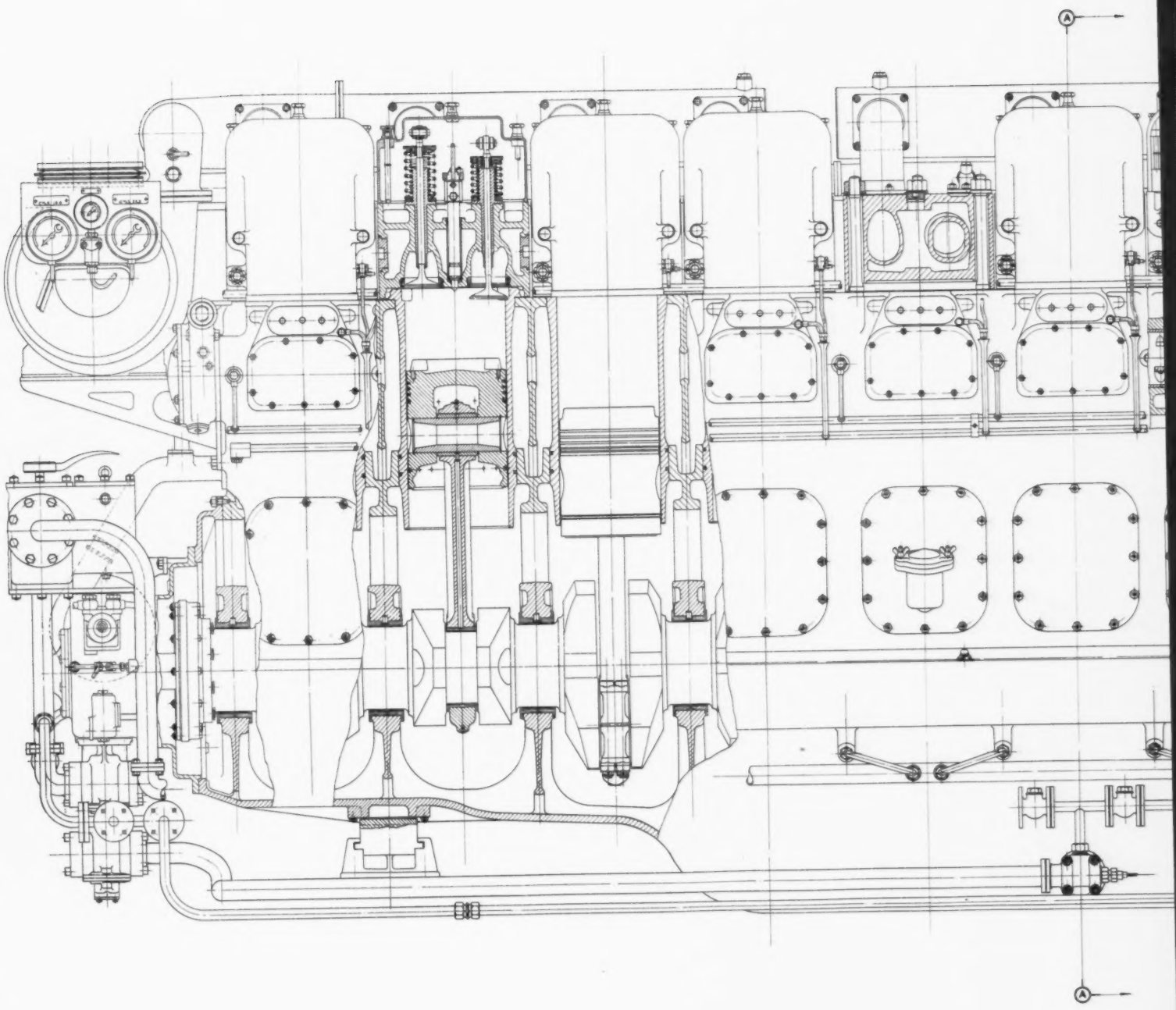
At the driving end of the crankshaft a split chain-wheel, accurately registered for endwise and angular locations, is clamped for driving by duplex chain a layshaft high up in the crankcase, whence by spur gears the camshaft is driven. The chain is tensioned by a movable idler sprocket, and means are provided for minute adjustment of angular relationship between the crankshaft and camshaft. The latter is a built-up shaft with valve and fuel injection pump cams and bearing journals assembled on appropriate lengths of shafting to permit easy endwise removal of the shaft from its bush-type bronze bearings, and quick replacement of any cam or bearing journal.

All the cam followers, which are of roller type, are carried in renewable guides mounted in the crankcase and, consequent on the high level of the camshaft, the push rods of the valve gear and the high-pressure fuel injection pipes between fuel pumps and injectors are unusually short. This gives great rigidity of push rods with minimum weight, and for fuel injection reduces harmful wave and compressibility effects in the pipelines.

In each cylinder head there is a central vertical fuel injection valve flanked by the air inlet valve and the exhaust valve. The air and exhaust valves are of poppet type, each loaded by two concentric coil springs with pitches of opposite hand, and the ports in which the valves are carried are integral with the cylinder head casting. In previous designs separate valve cages had been employed, but these restrict the aerodynamic design, and the use of special alloy seat inserts has enabled them to be dispensed with by improving seat conditions to such effect that no attention is required during normal periods between cylinder head removal for piston examination. The inserts are of alloy cast iron containing a high percentage of chromium, and are pressed into the cylinder heads. Each valve is operated by a lever of forged steel mounted in a cast bracket bolted to the cylinder head. The levers and push rods are clevis jointed and the latter are of tubular construction with ball and socket joints at their lower ends. At the valve-operating end of each lever the motion is transmitted through a freely-revolving roller. All the valve gear and fuel injection equipment is encased by light cast aluminium covers.

Lubrication and Cooling

Throughout the engine all moving parts are



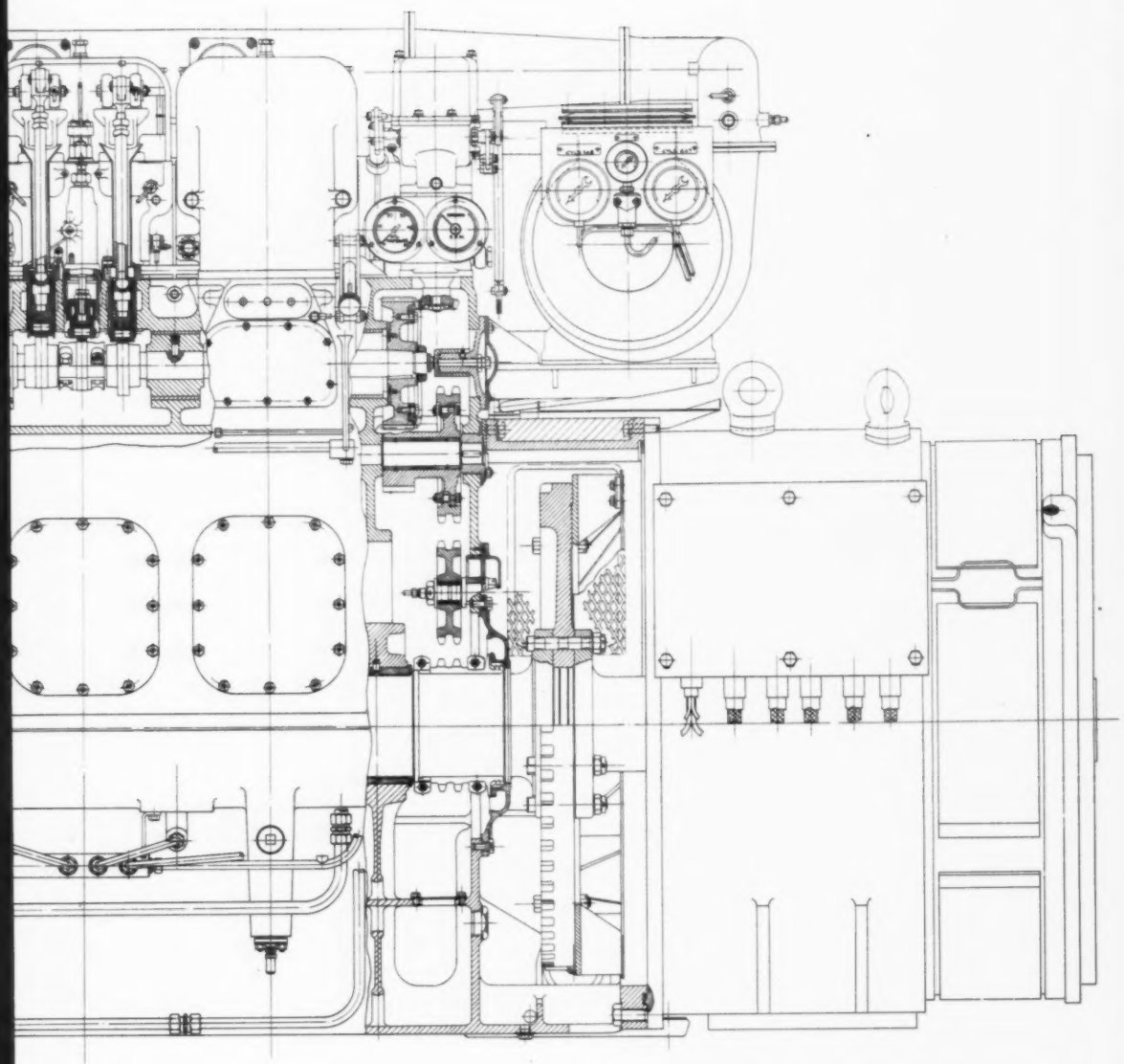


Fig. 4. Longitudinal section of '8SRKT' pressure charged diesel engine

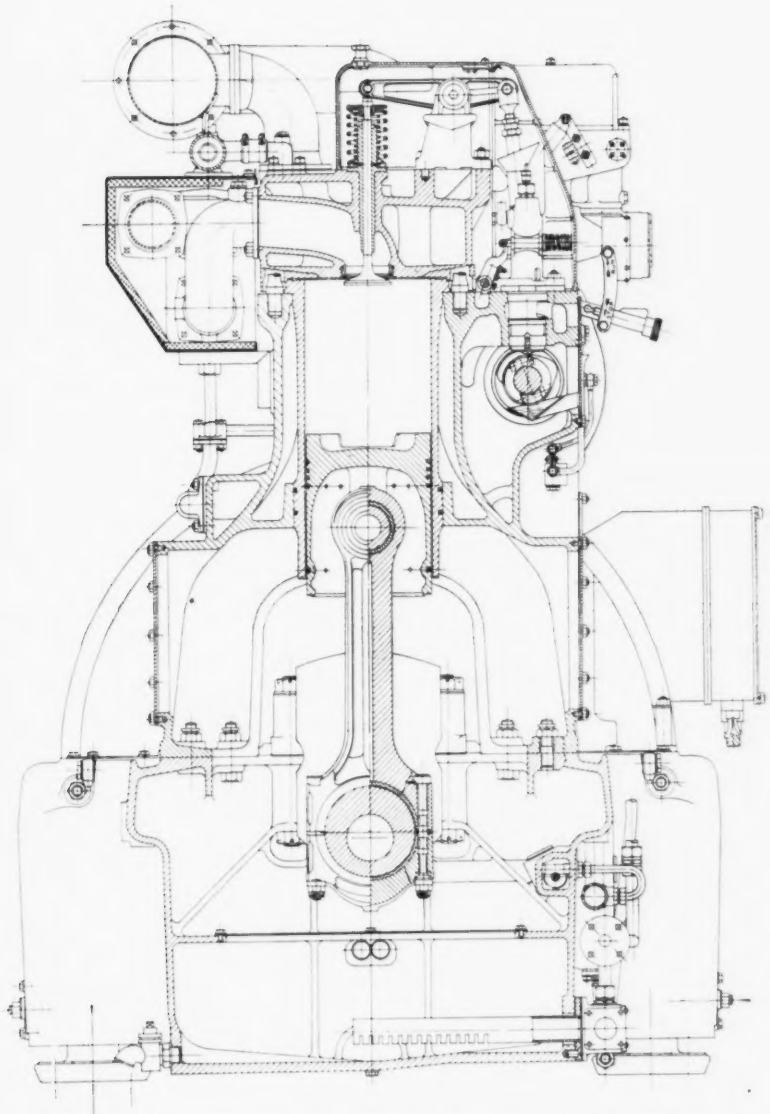


Fig. 5. Cross section, at 'AA' in Fig. 4, of '8SRKT' diesel engine

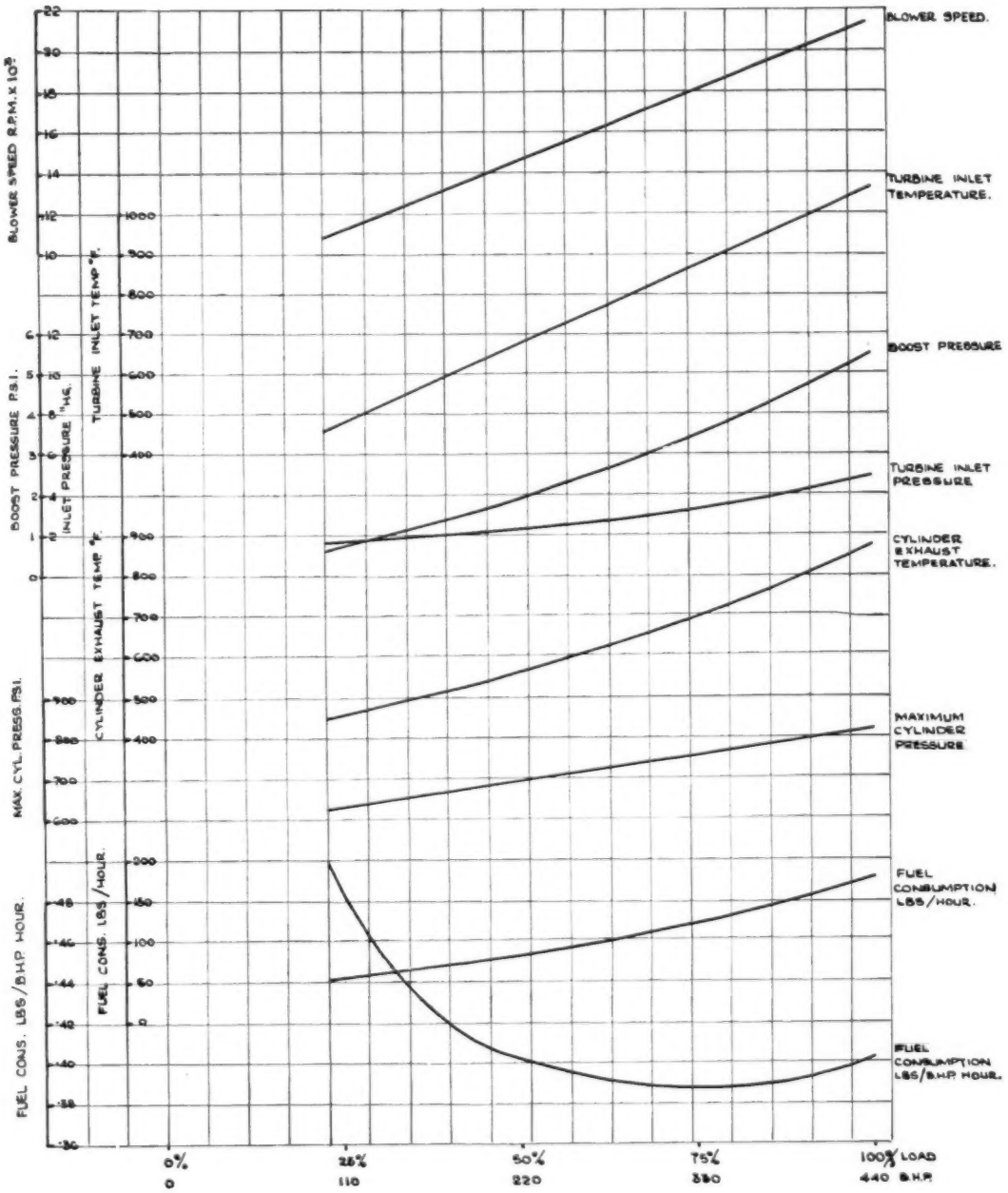


Fig. 6. Performance curves for '4SRKT' diesel engine

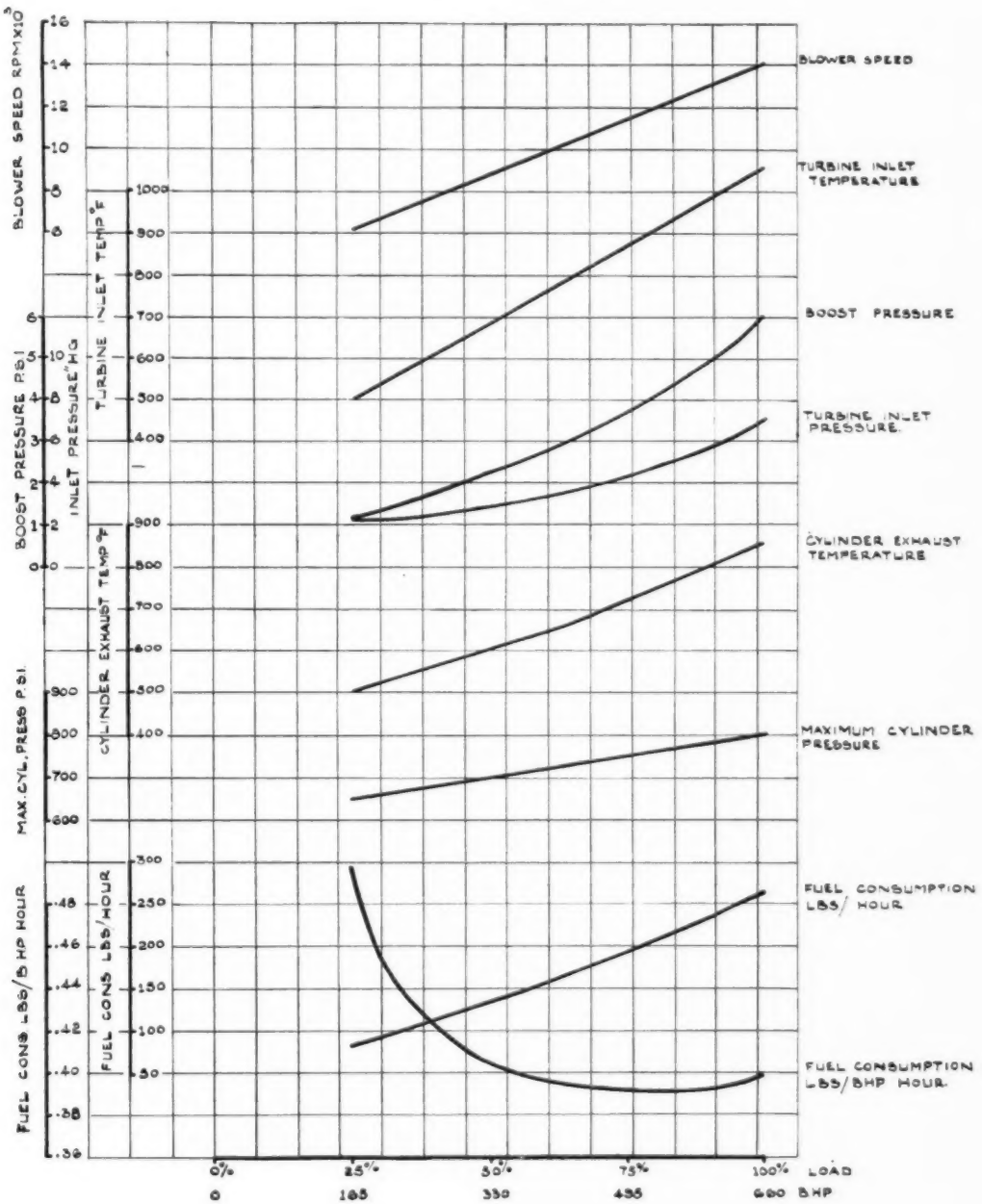


Fig. 7. Performance curves for '6SRKT' diesel engine

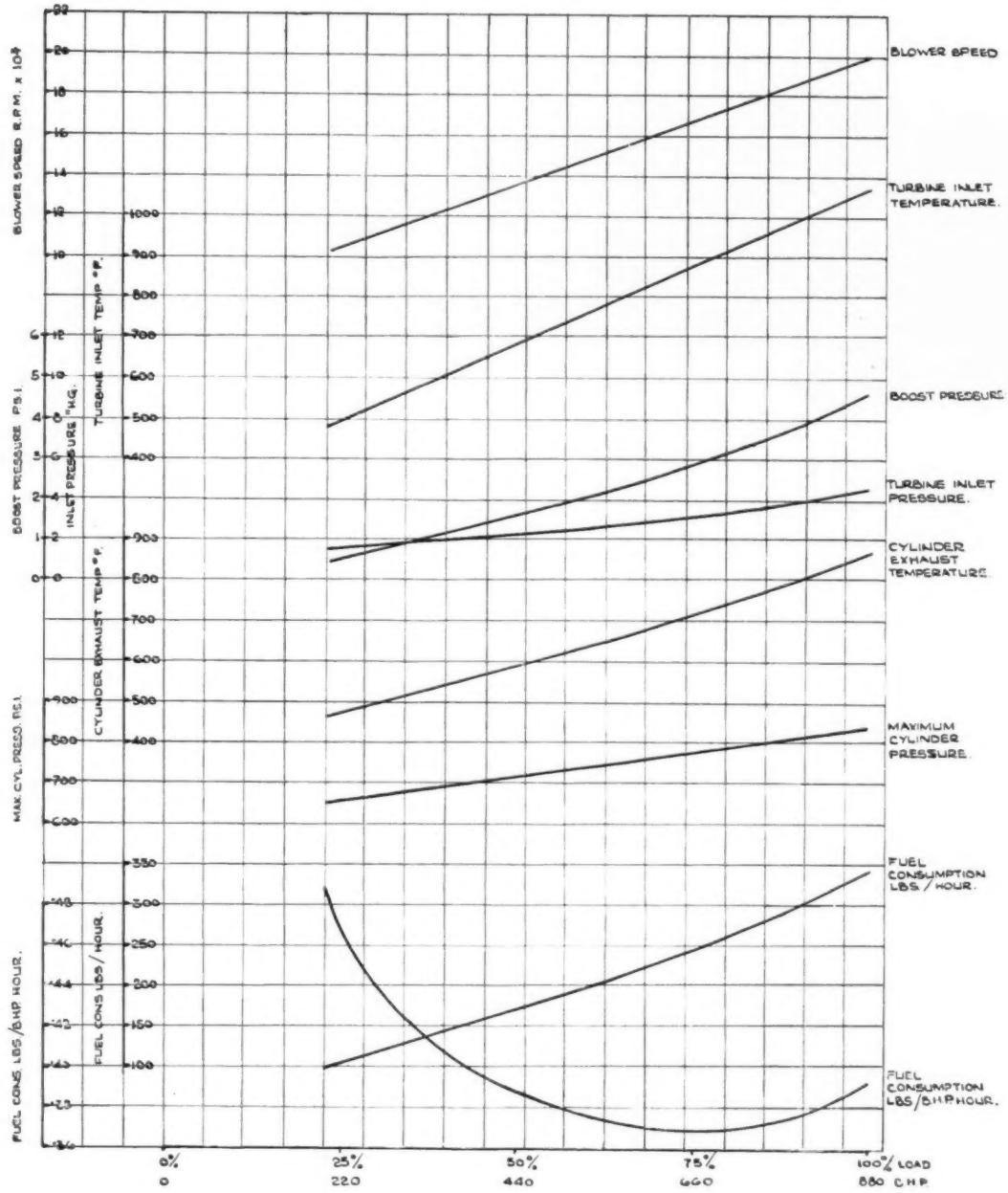


Fig. 8. Performance curves for '8SRKT' diesel engine

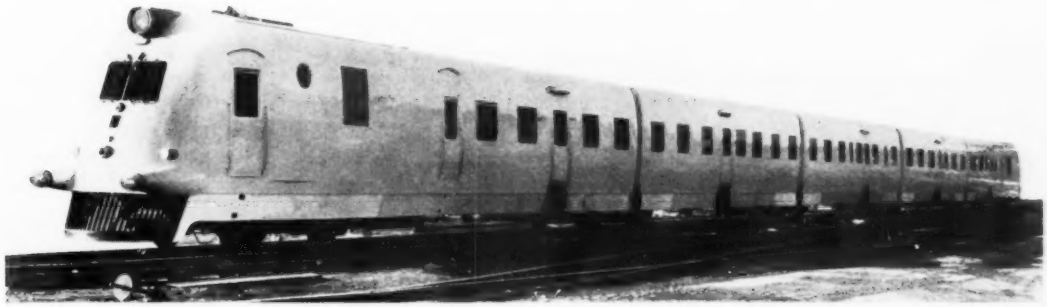


Fig. 9. An 800 h.p. 5-coach diesel-electric express train for the Egyptian State Railways

automatically force lubricated, a reduced pressure from that of the main system being employed for the valve gear components other than the camshafts. The oil pump, the fuel transfer pump and the cooling water circulating pump are mounted on the auxiliary drive casing at the free end of the engine and are driven from the end of the crankshaft through a resiliently connected layshaft and spiral gearing. The pumps have spigot locations in the drive casing and can be withdrawn and replaced without difficulty. For lubricating oil a duplex arrangement of gear pumps is used, the lower, larger pump circulating oil at low pressure through the radiator circuit, and the upper pump of the pair generating the high pressure necessary for engine lubrication. The system is equipped with suction strainer and non-return valve, differential and pressure relief valves and full-flow fine filters or strainers to clean the oil thoroughly before admission to the engine bearings.

The water pump employed is of centrifugal type, and from the end of its shaft a small positive pump is driven for lifting and/or pressurising the fuel supply to the engine.

Auxiliaries

Important auxiliaries are the main governor, of hydraulic servo type with electrically controlled speed settings and electric torque control on the full fuel setting, and the emergency governor. These are both driven from the camshaft, the former through bevel gearing at the flywheel end and the latter directly at the free end. The function of the emergency governor is, of course, to stop the engine if excessive speed occurs, and it does this by acting on the fuel injection pump control gear, turning the fuel pump plungers to the ' no delivery' position.

Another important auxiliary usually present in modern rail traction engine practice is the exhaust-driven turbo pressure charger, an independent machine mechanically, which derives energy from the engine exhaust gases and uses it to pressure-charge the engine. The pressure charger enables the engine output to be increased, and beneficially affects the thermal performance. It should be mentioned that different sizes of pressure charger are required to suit the differences in swept volumes. Thus the '4SRKT' engine uses a Napier TS100 pressure charger, the '6SRKT' is fitted with a Napier TS200 charger and the '8SRKT' with two Napier TS100 chargers.

Starting the engine in rail traction applications is accomplished through a special winding on the main generator, enabling it to be used as a motor with current supplied from a battery installed for this and other auxiliary electrical requirements.

Performance, Data and Service Examples

Performance curves for the three 'RK' type engines, showing fuel consumption, pressures, temperatures and supercharger performance over the full load range, are reproduced in Figs. 6, 7 and 8.

Further particulars of these engines are given in Table A. All three sizes are now in service in

considerable numbers and some typical applications have included five-coach and three-coach railcar sets for the Egyptian State Railways (Fig. 9) with two and one '4SRKT' units per set respectively, general purpose locomotives for the Tasmanian Government Railways (Frontispiece) with the '6SRKT' power unit, and switching locomotives with '8SRKT' engines for an Australian steel company (Fig. 10). Particulars of these trains and locomotives are given in Table B.

TABLE A

Particulars of 'RK' type pressure-charged rail traction diesel engines with 4, 6 and 8 cylinders.

Engine designation	4SRKT	6SRKT	8SRKT
Bore and stroke	10″ × 12″	10″ × 12″	10″ × 12″
Cylinder swept volume, cu. in.	3,770	5,655	7,540
Compression ratio	11.55/1	11.55/1	11.55/1
Operating speeds :—			
1st	450 r.p.m.	450 r.p.m.	450 r.p.m.
2nd	620 ,,	620 ,,	620 ,,
3rd	750 ,,	750 ,,	750 ,,
Full power	440 b.h.p.	660 b.h.p.	880 b.h.p.
Corresponding b.m.e.p.	123 p.s.i.	123 p.s.i.	123 p.s.i.
Firing order	1, 3, 4, 2.	1, 5, 3, 6, 2, 4.	1, 5, 7, 3, 8, 4, 2, 6.
Dimensions and weight, including main and auxiliary generators:—			
Overall length	13′ 0″	16′ 8″	17′ 9″
Overall width	4′ 3″	4′ 8″	4′ 8″
Overall height	7′ 0″	7′ 3½″	7′ 0″
Total dry weight	22,920 lb.	29,700 lb.	37,150 lb.
Crankshaft journal dia.	8½″		
Crankpin dia.	7¼″		
Gudgeon pin dia.	3½″		
Camshaft journal dia.	5½″		
Injection pump	16 mm. bore × 15 mm. stroke		
,, nozzle	6 holes 0.43 mm. dia.		
,, timing	10° – 12° b.t.d.c.		
Valve data :—			
Inlet valve opens	90° b.t.d.c.		
,, ,, closes	45° a.b.d.c.		
Exhaust ,, opens	45° b.b.d.c.		
,, ,, closes	65° a.t.d.c.		
Inlet ,, dia.	4.0625″		
,, ,, lift	1.11″		
Exhaust ,, dia.	3.8125″		
,, ,, lift	1.11″		
Cooling :—			
Water pump speed	1.94 × engine speed.	1.94 × engine speed.	1.94 × engine speed.
,, ,, capacity	4,800 g.p.h.	5,600 g.p.h.	7,500 g.p.h.

Applicable to all the three engine sizes.

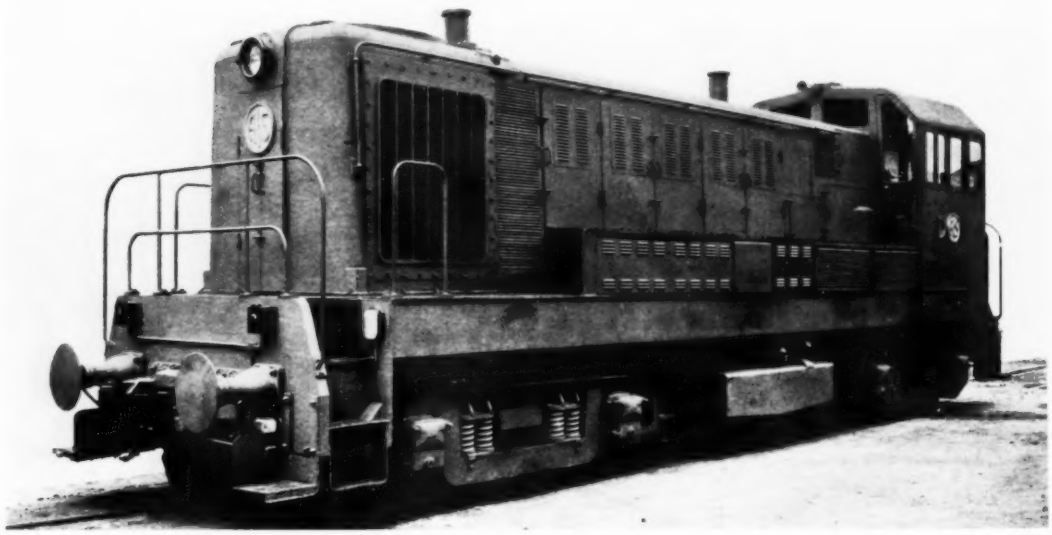


Fig. 10. An 800 h.p. diesel-electric shunting and transfer locomotive operated by an Australian steel company

Lubrication :—			
Oil pump speed ..	1.435× engine speed.	1.435× engine speed.	1.435× engine speed.
Scavenge pump capacity ..	1,480 g.p.h.	2,640 g.p.h.	2,640 g.p.h.
Pressure pump capacity ..	1,385 g.p.h.	2,500 g.p.h.	2,500 g.p.h.

Governing :—			
Governor speed ..	1.596× engine speed.	1.596× engine speed.	1.596× engine speed.
Temporary speed variation, not exceeding	10%	10%	10%
Permanent speed variation, not exceeding	4%	4%	4%

TABLE B.

Locomotive and train particulars.

Operator	Tasmanian Government Railways.	Egyptian State Railways.	Australian Steel Company.
Locomotive or train	General purpose locomotive.	5-coach 800 h.p. train.	Switching locomotive.
Illustration	Frontispiece	Fig. 9	Fig. 10
Track gauge	3′ 6″	4′ 8½″	4′ 8½″
Wheel arrangement	Bo-Bo	Articulated train	Bo-Bo
Overall length over buffers	36′ 8″	255′ 3″	41′ 0″
Overall width	9′ 5½″	9′ 6″	9′ 3″
Overall height	12′ 3″	12′ 6⅜″	14′ 0″
Total wheelbase	24′ 9″	*46′ 3″	30′ 6″
Rigid wheelbase	8′ 3″	8′ 7″	7′ 6″
Diameter of motored wheels	36½″	40″	44″
Diameter of unmotored wheels	—	34″	—
Weight in working order	55 tons	152 tons (laden)	87 tons
Adhesive weight	55 tons	62 tons (laden)	87 tons
Maximum axle loading	13.75 tons	16 tons	21.75 tons
Fuel tank capacity	340 gallons	400 gallons	600 gallons
Diesel engine type	6SRKT	Two 4SRKT	8SRKT
Diesel engine rating	660 b.h.p. at 750 r.p.m.	400 b.h.p. at 750 r.p.m.	800 b.h.p. at 750 r.p.m.
Main generator	6-pole EE827	Two 6-pole EE824	6-pole EE819
Auxiliary generator	6-pole EE906	Two 6-pole EE906	4-pole EE905
Traction motors	Four 4-pole EE521	Four 4-pole EE517	Four 4-pole EE506
Gear ratio	67/16	†59/17	‡18.6/1
Maximum starting tractive effort ..	31,000 lb.	†18,000 lb.	60,000 lb.
One-hour tractive effort	15,800 lb. at 12 m.p.h.	† 9,500 lb. at 22 m.p.h.	30,000 lb. at 6 m.p.h.
Continuous tractive effort	12,700 lb. at 15 m.p.h.	† 7,250 lb. at 29 m.p.h.	23,500 lb. at 7 m.p.h.
Maximum service speed	55 m.p.h.	†75 m.p.h.	21 m.p.h.
Number built or ordered	32	♂19	8

* Distance between bogie centres.
‡ Double reduction gearing.
† Data tabulated for express units. Suburban units have 65/16 gear ratio, 22,000 lb. starting tractive effort, one-hour tractive effort 12,000 lb. at 19 m.p.h., continuous tractive effort 8,500 lb. at 26 m.p.h. and maximum service speed 62 m.p.h.
♂ 10 express units and 9 suburban units.

New Engine Ratings

Since the introduction of the 'RK' engines development work has been proceeding at the Company's diesel engine research station, and 'Mark 2' versions of both the 'RK' and 'V' types are now being made for rail traction service having outputs, with pressure charging, of 125 b.h.p. per cylinder at 850 r.p.m.

Governors and Control Gear for Diesel Engines

By L. BERMAN, M.A., A.M.I.Mech.E.,
Assistant Chief Engineer (Preston), Diesel Engine Division.

THE PRACTICE OF fitting a governor to an engine in order to control its speed is not, of course, peculiar to diesel engines, and various types of governor were well developed in connection with the earlier steam and gas engines. Special applications and operating conditions have, however, necessitated the development of more complex governors for diesel engines, and it is proposed to discuss some of these in this article.

The earlier types of diesel engine were mostly built with comparatively few cylinders and had large flywheels in proportion to their power output, in order to achieve reasonable speed regularity. There was therefore no difficulty in controlling the speed by means of a straightforward centrifugal governor, which was usually of fairly massive construction so as to give sufficient power to operate the fuel control mechanism. In later years, however, both speed and the number of cylinders increased while the size of an engine of a given power decreased, so that it often became desirable to use governors provided with some means of rapidly exerting large forces, but small in size.

An obvious solution is to use a servo mechanism with lubricating oil under pressure as the operating medium. The oil can be taken from the engine system or the governor can have its own oil tank and pump. The servo mechanism can either be an adjunct to a normal governor, giving simply a multiplication of force, or it can be an inherent part of the governor system.

Servo Governor

A servo governor of the latter type used on 'English Electric' 'K,' 'RK' and 'V' type engines is shown in Fig. 1. The centrifugal element A is driven through the flexible coupling and moves the pilot valve B ; this valve controls the flow of oil to or from the cylinder C. If the load increases and the speed consequently falls, the weights move inwards and lower the valve ; oil is admitted to the cylinder and the piston D rises, thus rotating the shaft E which alters the fuel pump setting so as to inject more fuel. At the same time, the upward movement of the piston raises the end of lever F and makes it rotate slightly about roller G, so that the plunger H rises and reduces the compression of the spring. The system will therefore reach equilibrium again, with the valve closed, at a slightly lower speed than before. This slight fall in speed with increase in load is necessary for stability with any normal type of governor. Limiting figures for speed variation are laid down in a British Standard Specification.

If the load decreases and the speed rises, the pilot valve is raised and the servo cylinder is connected to the drain passages; the spring pushes the piston downwards and the fuel quantity is decreased.

The force available to move the fuel pump control gear, which must be considerable on, say, a 16-cylinder vee-type engine, can evidently be made large without the necessity for large rotating weights, etc. The weights are carried on needle roller bearings so as to minimise friction and make the governor sensitive. The engine speed can be adjusted by the handwheel J or by an electric motor controlled from the switchboard, if desired. By means of screw K it is possible to vary the amount of speed change for a given change of load, so that the governor of an engine can be made to have the same characteristic as that of other engines or turbines when several alternators are being run in parallel and are required to share the load equally.

It will be seen that the method of operation of the governor automatically provides two safeguards : (1) the engine cannot run until some oil pressure has been built up, and (2) the fuel is cut off and the engine stops if the pressure drops below a certain value.

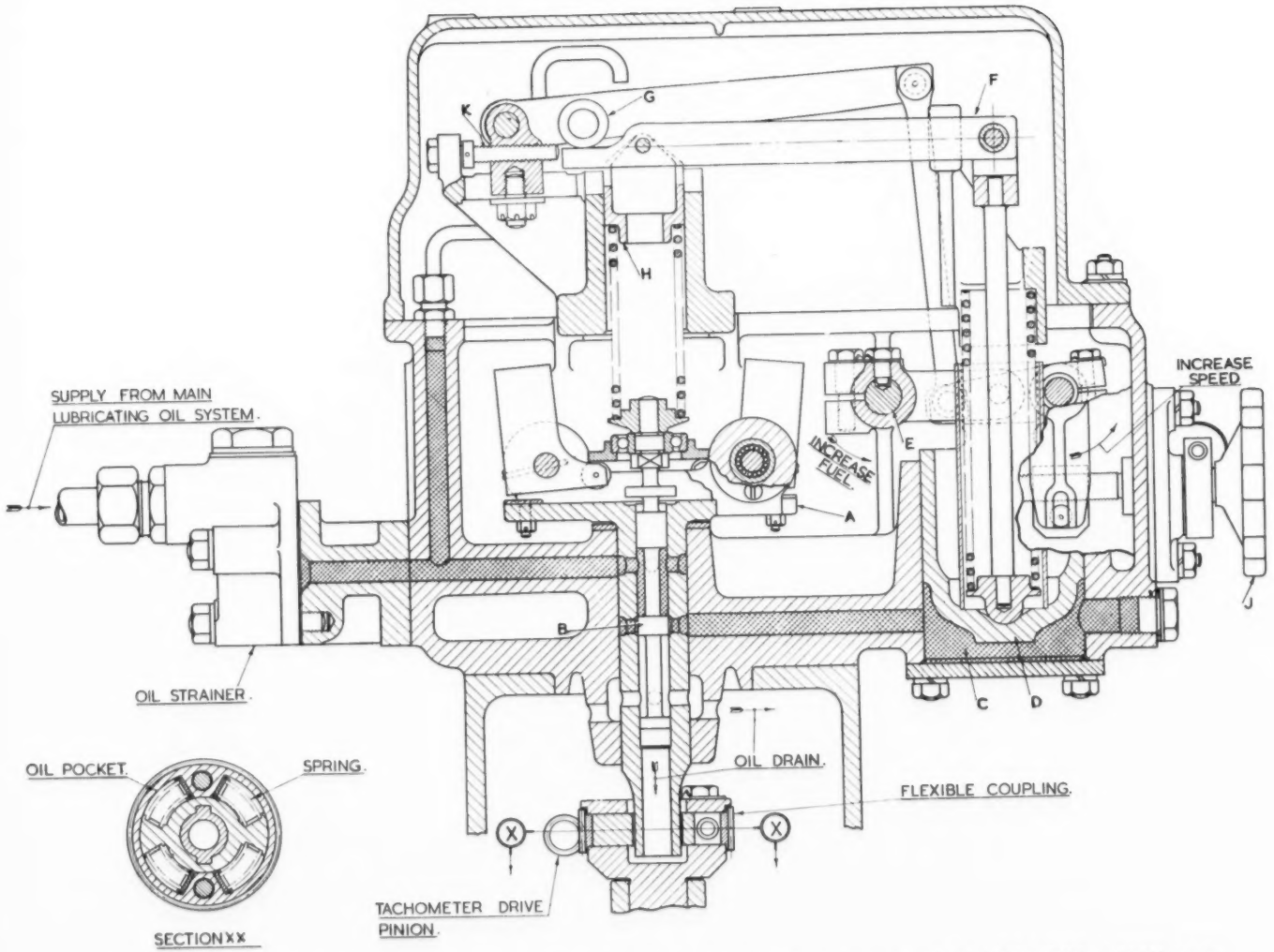


Fig. 1. *Sectional view of the servo governor used on 'English Electric' diesel engines of the 'K,' 'RK' and 'V' types*

The governor is connected to the fuel pumps as shown in Fig. 2. The fixed and free levers at each pump are held together by a spring and normally move as one unit. If, however, a pump should stick, these levers can separate and allow the governor to move all the other pumps in the direction which reduces fuel ; there is, therefore, no risk of the engine running away in such a case. This safety feature is incorporated in all ' English Electric ' engines which have separate fuel pumps. A spring-loaded link is fitted between the governor and the control shaft ; when the hand lever is used to stop the engine, this link is compressed and it is not necessary to use enough force to move the servo piston against the oil pressure.

The operation of a governor of this type has been thoroughly investigated by fitting suitable electrical pick-ups connected to a cathode-ray oscillograph ; the results are shown in Fig. 3, which shows the action when full load is suddenly thrown off an engine. It will be seen that the response of the pilot valve is very rapid and that the piston begins to move the fuel control after 0.2 second. The fuel is reduced to the no-load value in 0.7 second and the engine speed then gradually falls to the steady no-load figure. The temporary change of speed on throwing load on or off depends, of course, on the kinetic energy of the system as well as on the behaviour of the governor.

It was mentioned earlier that a flexible coupling is fitted to the governor ; this has four light springs which ensure a natural frequency well below the

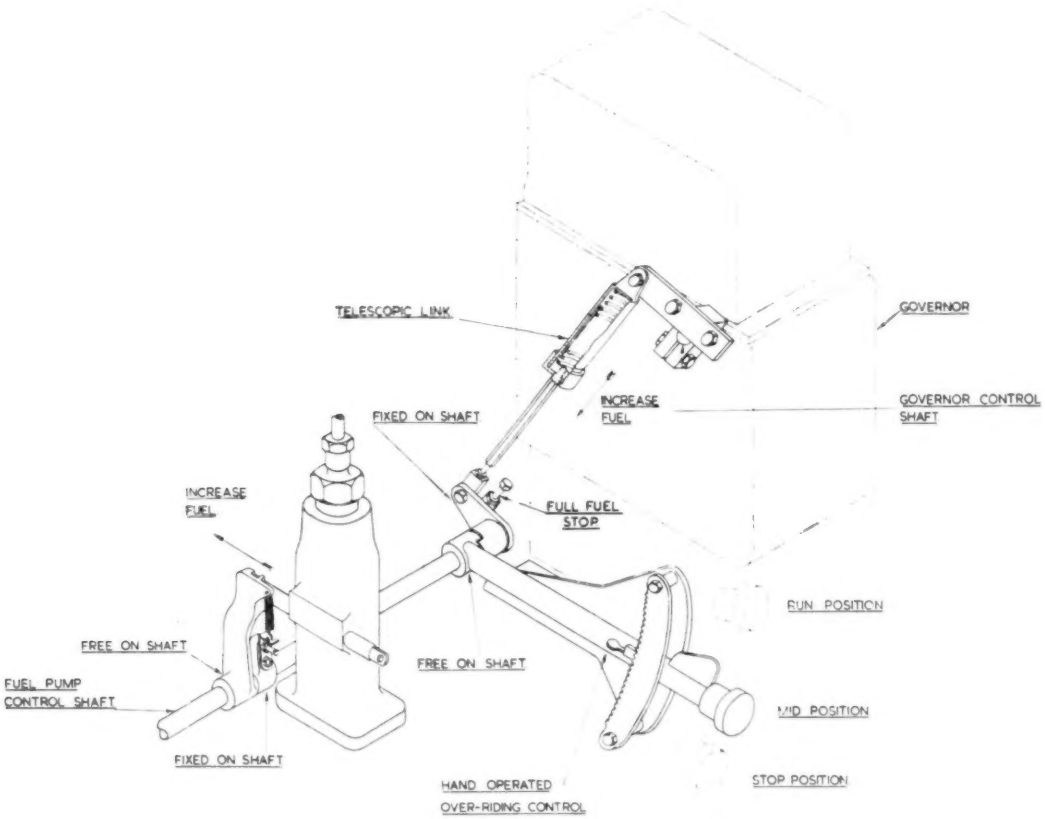


Fig. 2. Fuel pump control gear

frequency of the impulses from the engine at its lowest running speed. In order to prevent any building up of free oscillations, the coupling is filled with oil and relative movement between the driving and driven sides is damped by flow of the oil through small passages. Fig. 4 shows the angular displacement of this coupling on an 8-cylinder 4-stroke '8SK' type engine ; the impulses occur four times per revolution, which is the firing frequency. The motion of the governor itself was observed during tests to be quite uniform, showing that the flexible coupling was satisfactorily smoothing out the irregular motion of the driving shaft.

Isochronous Governors

Isochronous governors, which act to maintain exactly constant speed, can be constructed by means of systems arranged to give a temporary change of speed and then to alter some part of the mechanism so as to restore the speed. For example, the governor previously described was satisfactorily modified for experimental purposes, as shown in Fig. 5. The link connecting piston D and lever F is replaced by an internal piston M and piston rod N ; the end of lever F is connected to a spring O. When a sudden change of load occurs, pistons D and M move together at first, since the spaces in the servo piston are filled with oil ; hence the governor has temporarily the same speed-load relation as before, which ensures stability. Spring O then gradually brings piston M back to its original position by forcing the oil from one side of the piston to the other through a small hole ; the engine speed therefore returns to the original value. For small changes of load there is, in practice, no noticeable speed change. Although the idea of an isochronous governor is attractive at first sight, it must be realised that it involves additional complexity and there are cases where it is definitely undesirable.

Traction Control Systems

A field in which many special control systems have been devised is that of diesel-electric rail traction ; The English Electric Company has, of course, very wide experience in this work. There are two main requirements here ; the engine speed must be capable of control by the driver over an appreciable range, say 2 : 1, so that the required

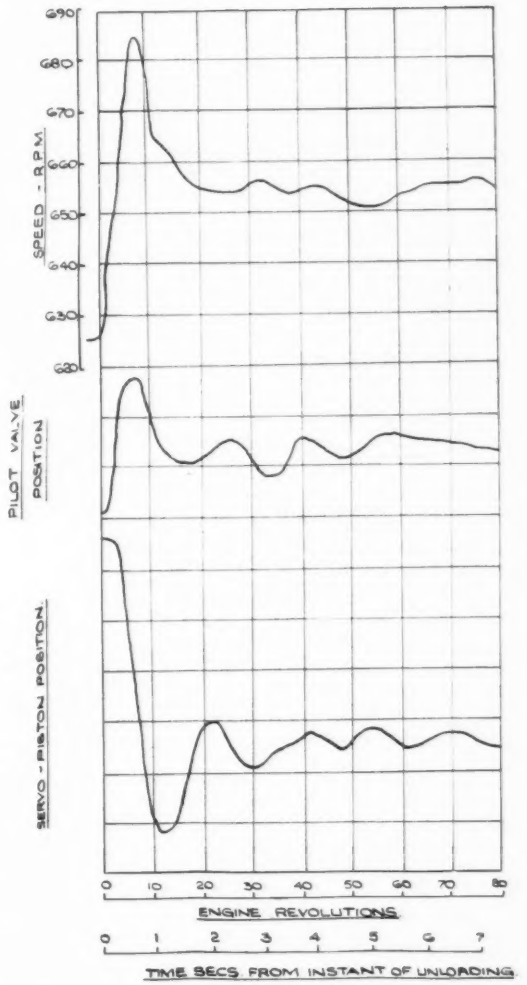


Fig. 3. Reproduced oscillogram of tests on servo governor, showing action when full load is suddenly thrown off an engine

output can be developed at any time without running the engine faster than necessary, and at the same time the electrical conditions should be regulated so that the engine is made to produce its full horse-power when desired, without any risk of overloading and stalling.

The power input needed by the generator depends on the weight of the train, gradient, speed, etc., so that it is not possible to ensure the

use of all the available engine power in varying conditions without some automatic apparatus to adjust the generator characteristics. The usual method employed is some form of field current control operated by the governor which, when full power is demanded by the train, keeps the generator output at a value that corresponds to full engine power. A typical generator characteristic curve for such a system is shown in Fig. 6. At low train speeds the current must not exceed a certain value, giving a point A ; as the speed increases, the current decreases and the voltage is made to increase in such a way that the curve is a hyperbola of constant power. Eventually point B is reached, at which the field current is at its maximum. At any point between A and B, the engine is working at its full power.

The parts of a control system of this type as used by The English Electric Company are shown in Figs. 7 and 8. The governor is generally similar to that previously described, but it is arranged to

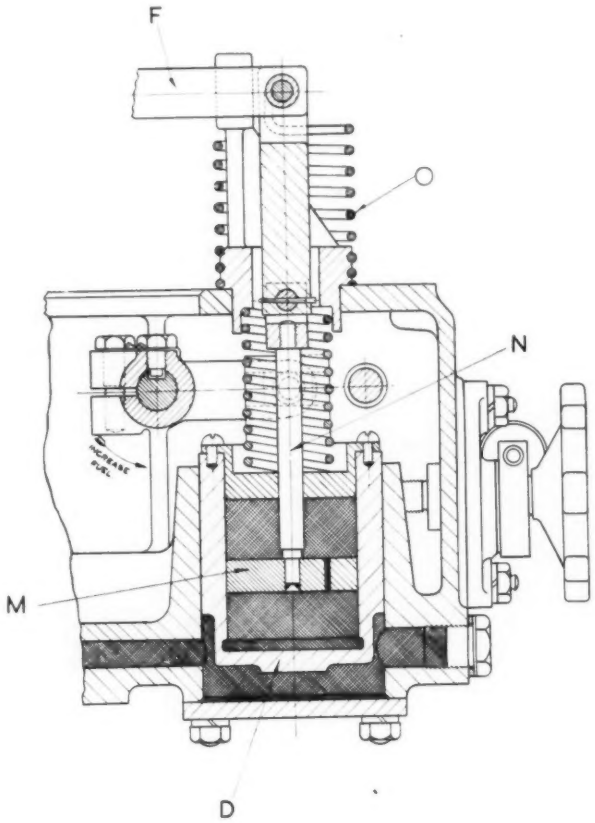
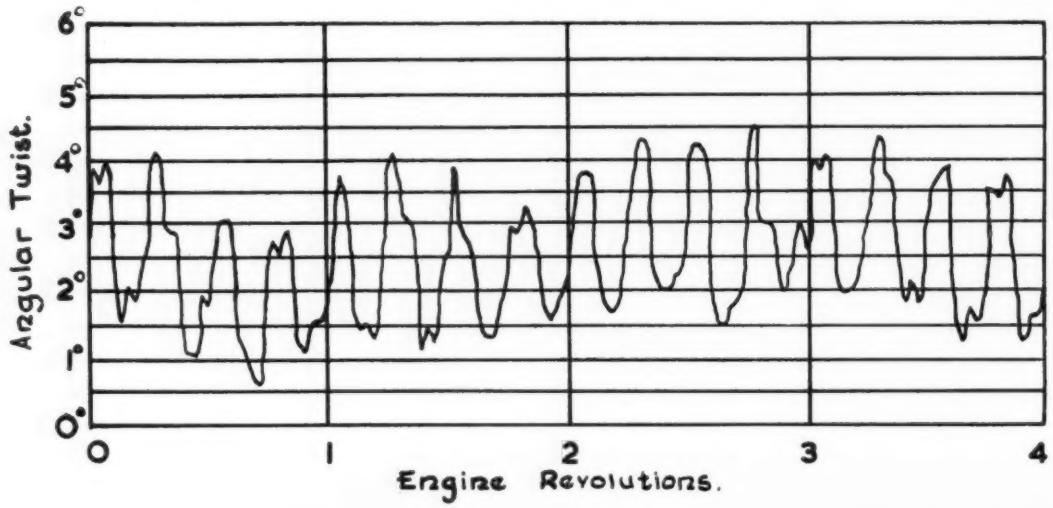


Fig. 4 (*below*). *Twist in flexible coupling of governor on an* 8*-cylinder* 4*-stroke* '8*SK*' *type engine*

Fig. 5 (*right*). *Section of isochronous governor*



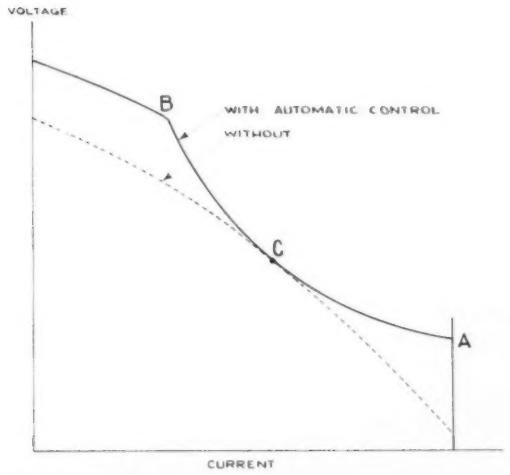


Fig. 6. Typical traction generator characteristic curves

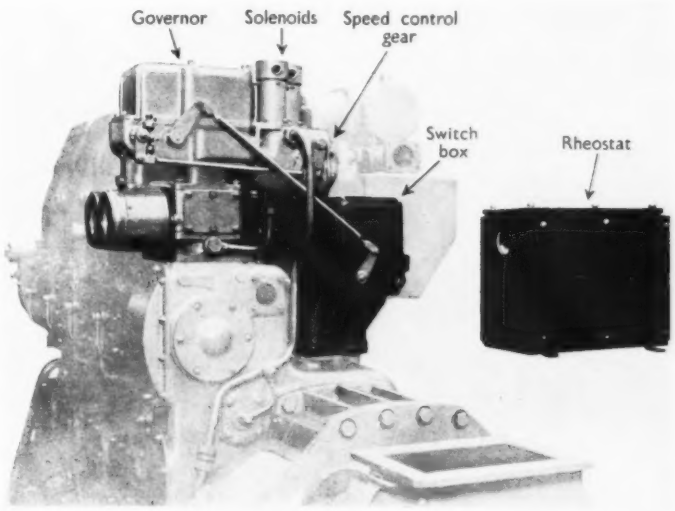
cover a wide speed range and the speed setting is varied by oil-operated pistons controlled by solenoids. The governor piston is connected to the fuel pump control gear and also to two switches. The latter are in the open-circuit position when the governor is at the full-fuel position. If the engine is developing full power and the conditions change so that the generator demand increases, the speed falls and the governor raises the piston, but the fuel quantity does not increase, since the control linkage comes up against the stop ; the spring-loaded link allows the governor to move. One of the switches closes and this starts the rheostat motor, which rotates in such a direction as to reduce the current in the generator field coil. When the generator demand is brought back to the full-load value, the engine speed rises again and the switch opens, so that the rheostat comes to rest in a new position. If the generator demand decreases, the second switch is closed by the governor and the rheostat movement is reversed. The rheostat can be mounted in any convenient position, not necessarily near the governor. In practice, the rheostat is constructed with a large number of switches which are closed in turn by a moving roller, as shown in Fig. 9.

It will be seen that this process is, in effect, governing by electrical means and it is complementary to the normal governing by fuel variation. Since the determining factor is engine speed and not kilowatt output, any variations in engine tune are automatically taken care of and the engine is kept just to the full fuel setting, so that it cannot be overloaded. The engine speed varies to some extent in the electrical governing range, but this variation is quite small. It is possible to add a mechanism to make the maximum fuel quantity different at the various speed settings, so that the torque-speed law follows any desired curve.

When two locomotives are operated in multiple unit with this type of control gear, it is only necessary to arrange for the two governors to be linked to the driver's handle as far as speed setting is concerned ; each engine is automatically kept to its proper torque and the two power units are independent, with no connections carrying heavy currents.

For some applications the slight additional complexity involved in this automatic control is

Fig. 7. Governor and associated equipment for diesel-electric traction control system



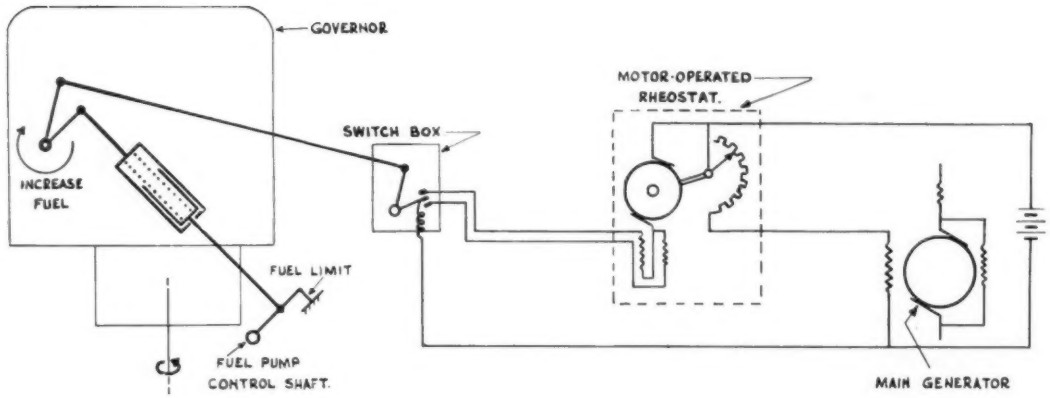


Fig. 8. Diagrammatic arrangement of control system for diesel-electric traction

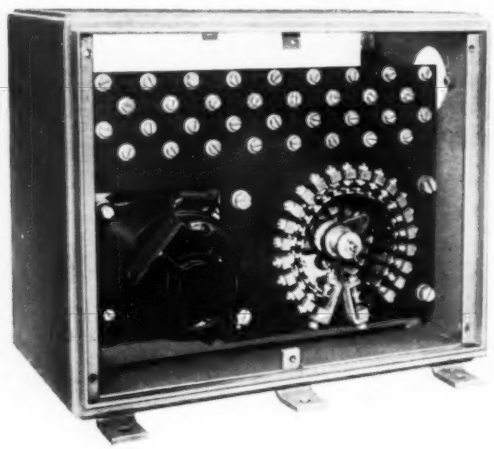


Fig. 9.—Rheostat for diesel-electric locomotive control system

not justified by the service conditions, and the nearest approach to the full-power curve without automatic control is shown by the dotted generator characteristic in Fig. 6. Full power is reached only at point C. To obviate the risk of the engine stalling, if its output should be reduced by some minor defect, and to allow for the greater generator output when the field coils are cold, the normal generator output is adjusted to a few per cent. below the dotted curve.

Close collaboration between the mechanical and electrical designers is obviously necessary in the development of control systems such as those just described.

Numerous other problems connected with governing can occur and need appropriate solutions, but this article has concentrated on matters of general principle and wide application.

The Manufacture of Asbestos-Cement Materials

ASBESTOS HAS BECOME renowned as one of the most useful minerals known to man since the commencement of the industrial age. For many years its use was confined to situations where its fire-resisting properties could be applied to greatest advantage but in recent times, when associated with cement, it has provided the foundation of an important material employed by many sections of the building industry—asbestos-cement sheeting. This is used universally for the roofing and side sheeting of factories, hangars, garages, and many other types of building and, when moulded into shape, provides the only known substitute for metal for use in gutters, rain pipes, soil pipes, flues and for numerous other purposes in the building industry.

One of the principal factories in Great Britain where this class of material is produced is that of the Atlas Stone Company at Meldreth, near Cambridge. The weekly output of some hundreds of tons takes the form of sheeting, both flat and corrugated, moulded pipes, bends, ventilators, and many other shapes. Asbestos-cement can be supplied in a wide range of colours. It has been exported in large quantities to all parts of the world.

The factory relies entirely on electricity for its motive power, and The English Electric Company provided most of the electrical equipment. Continuous operation of the manufacturing process takes place from Sunday night until Saturday morning each week, and any breakdown in the electrical plant would have serious consequences, since the liquid asbestos-cement mixture would set hard in the various stages of the process, involving many hours of maintenance work in breaking out and re-starting.

Raw Materials

The raw materials used in this industry are few

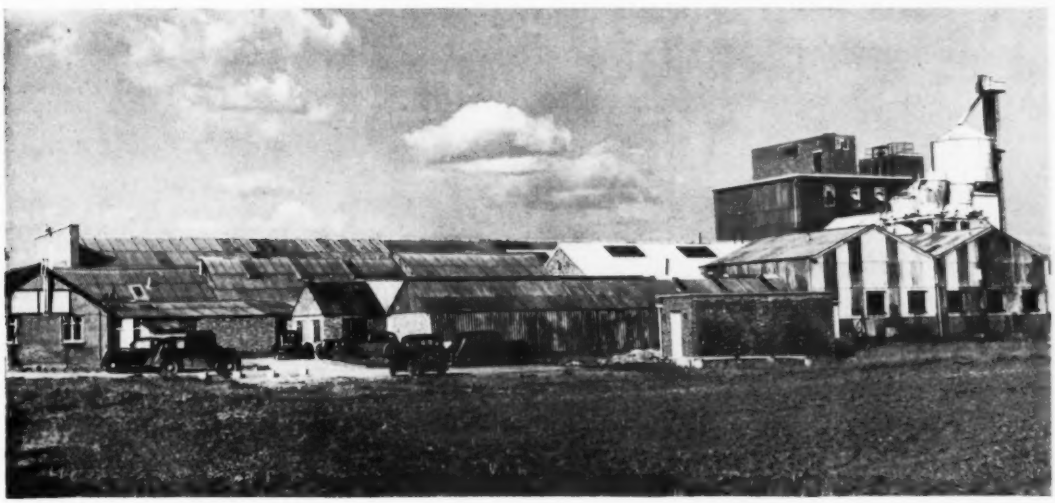


Fig. 1. A general view of the Atlas Stone Company's Meldreth Works for the manufacture of asbestos-cement materials

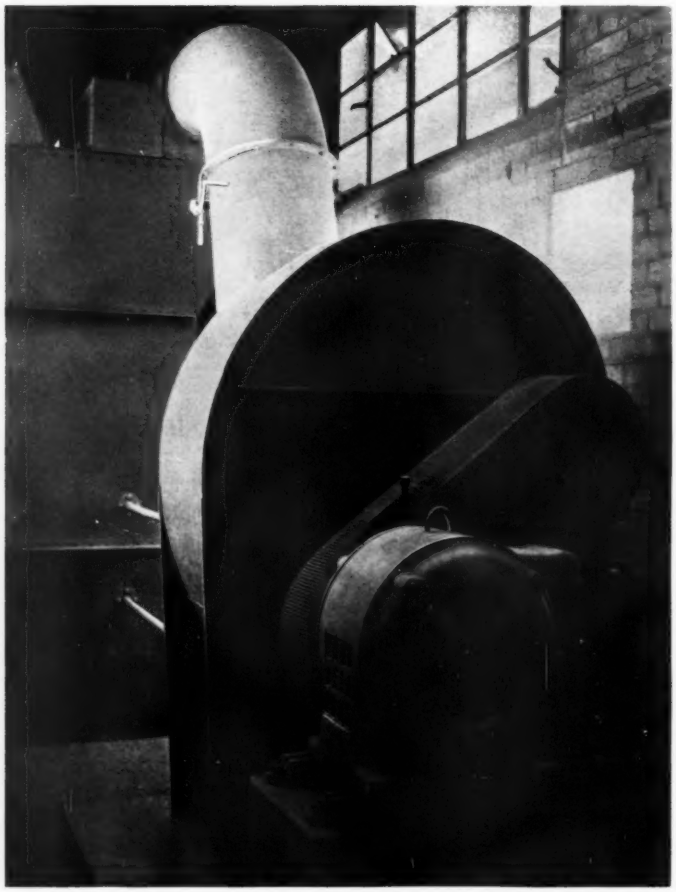


Fig. 2. Electric motor driving a fan associated with the dust collecting plant

in number and simple in character. They comprise asbestos, cement and water ; no binders, fillers, pigments, or other added chemical materials are used.

The asbestos is imported, limited supplies being available from Canada and South Africa. It comes to the factory in sacks in the form of loose fibres, each individual strand being of the order of $\frac{3}{4}$ in. long.

The cement is unloaded from lorries into a pit, whence it is lifted by an elevator to a silo. The drive for the elevator is provided by twin 5 h.p. motors which operate in the open air and under extremely dusty conditions.

The water is obtained from an artesian well on the premises.

Distribution of Cement and Asbestos

From the silo, the cement is handled by a screw conveyor, driven by an electric motor, which carries it to the weigher and distributing apparatus on an upper floor. Here an automatic mechanism initiates a number of operations by means of contactors which are equipped with time-delay relays where required. Solenoids operating mechanical levers ensure that the proper quantity of cement is weighed-out for each charge, and that when it is time to empty the weighing bin, the screw conveyors which carry the cement away have had time to discharge their loads before the next operation takes place. The cement then reaches large vats in which paddle shaped beaters are constantly revolving, and where the asbestos and cement are mixed.

Dust extraction plant is employed not only to rid the atmosphere near the machines of cement particles, but also to conserve a considerable quantity of cement which might otherwise be lost. It comprises five ' cyclones ' situated on an upper floor and fed by ducting from various machinery. Fans, powered by electric motors, drive the cement-laden air to the ' cyclones ' where the solid particles are deposited. The cement falls into bags and is available for re-use.

The asbestos fibre is taken by a conveyor to a disintegrating plant on a lower floor. A roller mill breaks up the fibre and ensures that it is in the " fluffy " state required, and it is then moved by a conveyor to a drying cylinder whence it is passed to a weigher where the exact amount required for each charge is ensured, again by automatic means. When the weigher has received its required quantity, auxiliary switches release solenoid-operated valves and the charge is thus directed to the mixers on the floor above. The requisite quantity of water is

automatically added by the opening of more solenoid-operated valves, and so the three ingredients of the asbestos board are brought together —cement, asbestos fibre, and water.

Mixing and Manufacturing Processes

For some 20 minutes the mixing process proceeds, the paddles being driven by totally-enclosed electric motors through Vee-belts from a floor below, and a certain amount of heat is generated due to the slaking of the cement. The charge is then removed by opening a valve, and proceeds by gravity to the stuff chests on the ground floor. These are large vats fitted with paddles which revolve slowly and keep the mixture from settling ; at this stage it is of a thinner consistency than might generally be expected, being no thicker than cocoa. From these chests the mixture is led away by ducts to the actual board-making machines on the same floor.

The manufacturing process is to some extent similar to that of paper-making. Large cylinders revolve in vats containing the mixture, the lower half of the cylinders only being in contact with the fluid. Across the top half of the cylinders runs a belt about 8 ft. wide composed of cotton and wool, and as the cylinders turn in the fluid they pick up the solid matter held in suspension and deposit it on the belt in a series of very thin layers which are continually accumulating.

At the end of its travel the belt moves in contact with a large " making " cylinder, and the layers of asbestos cement deposited on the belt are gradually taken off by pressure against this cylinder, which itself thus becomes coated with an even layer of

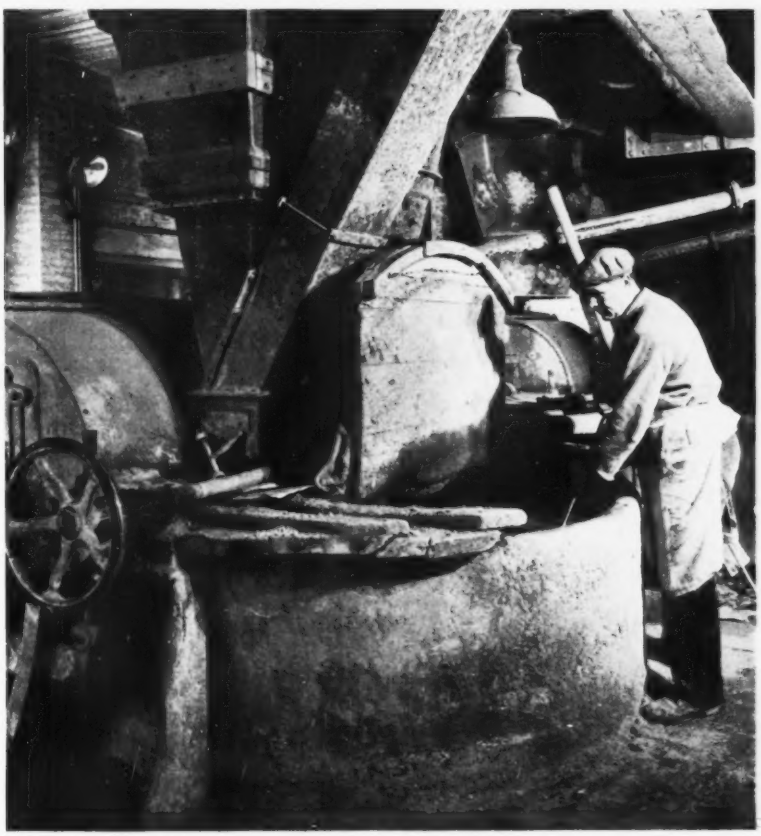


Fig. 3. The mixing floor where the three raw materials—asbestos, cement and water—are mixed in vats in which 'beaters' are continuously rotated. The drive is provided by the totally-enclosed motors seen in Fig. 4

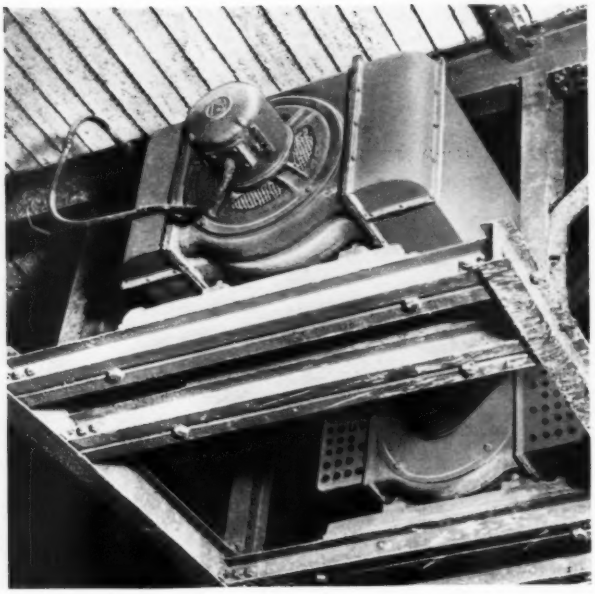


Fig. 4. 40 *h.p. totally-enclosed motors are used with Vee-belt drive to power the 'beaters' on the mixing floor*

the mix. A bell warning device informs the operator when the requisite thickness has accumulated. By a simple device the operator, while the cylinder is revolving, cuts the layer of mix, and the free end is drawn away on to a flat belt conveyor and is thus removed cleanly from the machine. The formation of one sheet takes only a few minutes, depending on the thickness required.

At this stage the material is in a flat green sheet about 8 ft. wide, the length varying as required. It is soft enough to be easily indented by the finger nail, and is pliable for any kind of moulding operation. In this state the sheets are immediately taken away to be moulded to the required shapes.

Moulding to Shape

This operation consists of pressing the wet material between steel sheets formed to the exact shapes required. One of the most common of these is the corrugated sheet, so frequently seen in use on roofs. The steel former has the corrugations impressed on its surface, and the asbestos-cement sheeting is pressed into the corrugations. The moulding time is only a matter of seconds. Other sheets are shaped by means of formers into tubes,

Fig. 5. *Windows being inserted in roofing sheets*

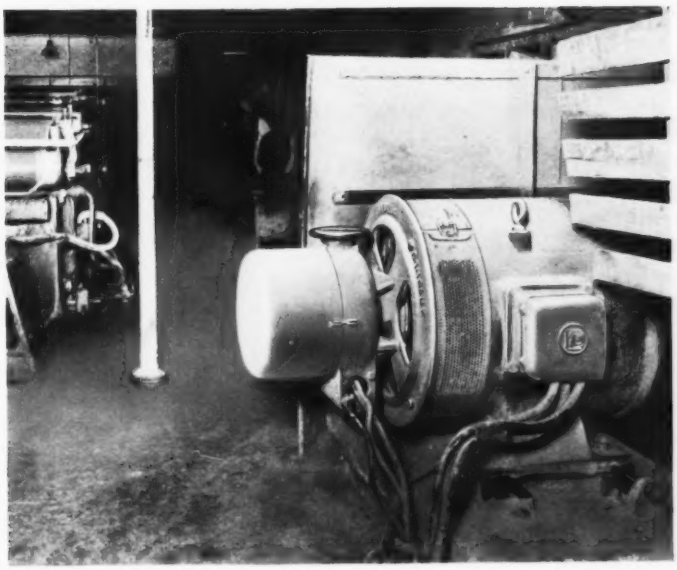
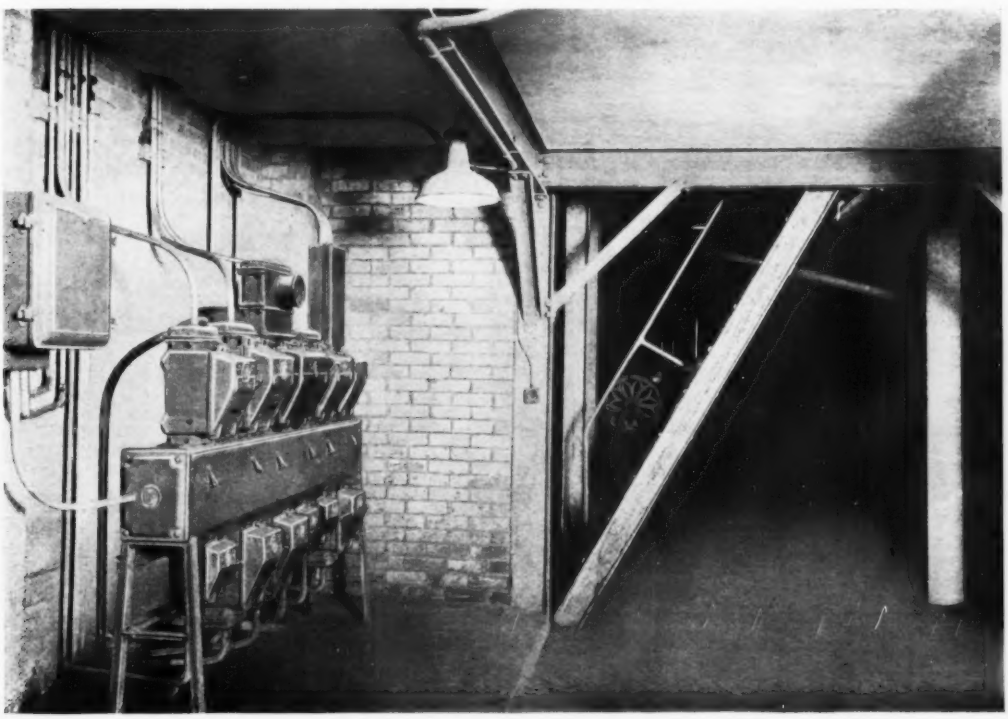


Fig. 6. This 160 h.p. motor was installed in 1929 and has never been out of commission

Fig. 7. For distributing power throughout the factory, a number of 400 volt 3-phase 'Combination' fuse-switchboards are used



gutters, angle-pieces, roof tiles and ridge-pieces, and domestic products such as draining boards.

No moulding is necessary, of course, when the final product is simply a flat sheet. In this case the sheets are merely removed from the machine and left to cure. In a very short time they achieve a degree of hardness which enables them to be handled, and in two or three hours they are reasonably rigid. After a period of about three weeks they gradually acquire the white appearance characteristic of this type of product, and after six or eight weeks become extremely tough and resistant to shock. They can then be cut, sawn, or drilled as may be required, while surface processing may be carried out in order to reproduce coloured marbling or to provide plain coloured surfaces.

Electrical Equipment

For the earlier machinery of the kind used, a single source of motive power was installed which drove all the plant by means of belts and countershafts, so that every part of the process operated in correct speed ratio. The substitution of electric drives had very many advantages, but it could not have been contemplated by the factory's management had they not been reasonably satisfied that no part of the electrical equipment would fail and, by being out of commission, stop the whole process. Having installed electric power, production duly continued without interruption, and one 'English Electric' motor of 160 h.p. has been operating since 1929 without ever being out of commission. This motor is subjected to cement and asbestos dust which is inevitable in factories of this nature.

In such installations as this, not only must the driving motors be robust and reliable, but the switchgear and other components of the electrical system must have an equal degree of reliability. 'Combination' fuse-switchgear, installed in conveniently situated panels throughout the factory, has proved capable of operating continuously without maintenance in dusty atmospheres.

Amongst other equipment associated with the main process is that for operating the artesian well from which the water supply is obtained. Compressed air is provided by two compressors, powered by electric motors, and is blown down a 2 in. air pipe inside a 6 in. water pipe. The resulting pressure at the bottom of the well forces the water upwards in the annular space between the pipes.

The modern canteen is electrically equipped, and 'Combination' fuse-switchgear is used to control the various circuits.

Acknowledgments

We are indebted to Mr. Bernard Davis, the Managing Director of the Atlas Stone Company, for permission to visit this factory and take the photographs which accompany this article ; and to Mr. Burgess the Manager, and Mr. May the Maintenance Engineer, for the facilities which were afforded to us on the occasion of our visit.

Design and Operation of Exhaust-Driven Turbo-Compressors for Diesel Engines

By A. J. PENN, M.I.Mech.E., Chief Engineer, Aero Gas Turbine Division, D. Napier & Son Limited.

This article is based on the paper read by the author in Paris on 8th May, 1951, before the International Internal Combustion Engine Congress, with whose kind permission it is published in this Journal.

IN RECENT YEARS the demand for four-stroke diesel engines of increased output for stationary, marine and traction purposes has resulted in the establishment of the turbo-charged diesel engine. Supercharging by means of the exhaust-driven turbo-compressor allows increased power output to be achieved without increase of engine capacity, and at the same time with improved overall thermal efficiency.

Diesel engines of the present day are operating continuously with a 50 per cent. increase in power output by supercharging to between 4 and 5 lb/sq. in. pressure, and increases in power output of over 100 per cent. are being obtained on development by supercharging, but in this case charge-cooling is employed. In the future, turbo-charged four-stroke diesel engines should be operating in service at twice the power output of normally aspirated engines of the same capacity, and with lower specific fuel consumption. To meet the present requirements of the diesel engine industry, exhaust-driven turbo-compressors with a maximum continuous rating of 1.5 : 1 compression ratio are supplied, and for future requirements these units have been developed for a maximum continuous rating of 2.0 : 1 compression ratio.

This article describes the design and operation of such turbo-compressors for four-stroke diesel engines, and discusses the problems associated with matching these units on the engines to obtain optimum performance.

General Description of Turbo-Compressor

A type of turbo-compressor suitable for universal application in the diesel engine industry has been designed and a series of basic units established to cover the wide range of engine outputs. This type of turbo-compressor comprises a single-stage centrifugal compressor driven by a single-stage

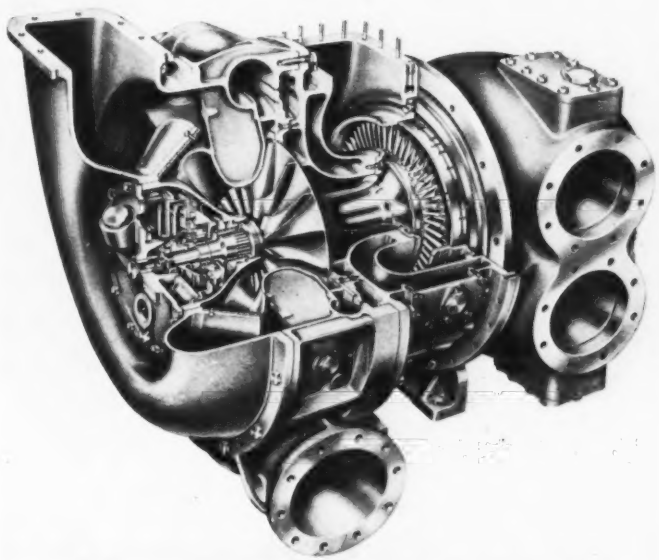


Fig. 1. *Sectional view of turbo-compressor unit*

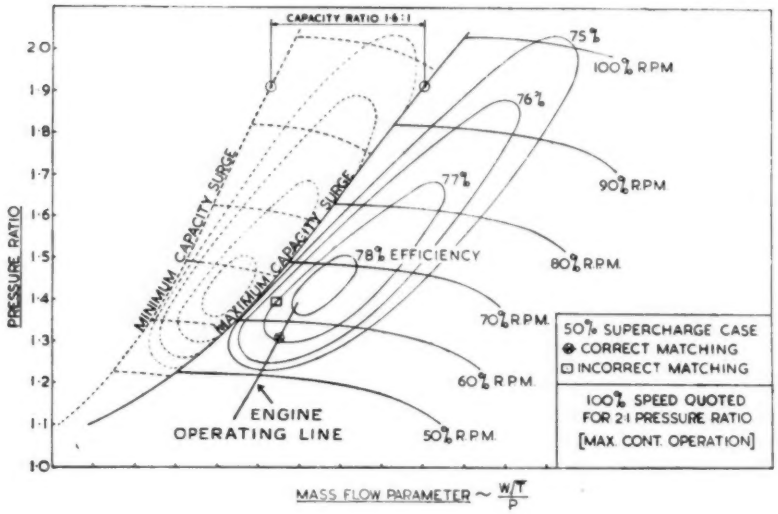


Fig. 2. Typical compressor characteristics for a basic unit

axial-flow turbine, the rotary components being mounted on a common shaft. A sectional view of the unit is shown in Fig. 1.

To cover the range of outputs required for sensitivity of matching to ensure optimum engine performance, a number of basic sizes is chosen each having the maximum capacity range possible with one impeller and various diffusers. Each basic size has a ratio of maximum to minimum capacities of 1.6 : 1, which has been determined as a result of previous research and is illustrated in Fig. 2.

The turbine is designed so that its capacity can be matched to the minimum and maximum rated compressor capacities for a given basic size, to ensure the combined unit operating at the maximum overall efficiency without causing compressor surge, this being achieved by designing the nozzle assembly and rotor blades in such a manner that the blade height can be adjusted in standard increments.

To suit engines of different design, the compressor and turbine inlet casings are made so that they may be positioned at any angle to the turbine outlet casing which in turn may be set at any angle relative to the mounting feet. By this means any arrangement of engine induction and exhaust manifolding can be accommodated. The mounting feet are adjustable to allow the turbo-compressor to be installed in a position best suited to the engine.

For engines with different numbers of cylinders but of the same capacity, three types of turbine inlet casing with two, three and four entries are designed for each basic size ; hence, by exhausting one, two or three cylinders into each manifold, one turbo-compressor can be employed for engines having from three to twelve cylinders.

From the mechanical aspect the unit has been designed for long life, with ease in servicing and in the replacement of parts, and has its own lubrication system. The turbine casings are designed for water-cooling by fresh or salt water.

The impeller and turbine wheel are mounted on a common shaft carried in two resiliently mounted ball bearings. The bearings are located outside the impeller and turbine wheel to facilitate inspection and replacement *in situ*. Each bearing has self-contained lubrication by means of a disc type oil pump mounted on the end of the shaft. Pressurised air sealing is incorporated in the rotor shaft assembly to ensure that the oil leakage from the bearings is an absolute minimum.

The compressor casings are made of sea-water-resisting aluminium alloy, and the impeller is machined from an aluminium forging. The turbine inlet and outlet casings are made of

heat-resisting cast iron. The turbine wheel comprises a forged disc fitted with blading having fir-tree root type fixings which are a sliding fit in the disc, thereby ensuring a high degree of vibration damping. The disc and blades are manufactured of high temperature steel.

The turbine wheel is stressed for continuous operation at a turbine inlet temperature of 600° C. and for operation at 650° C. for one-hour limited periods, relatively low stresses being employed to give long service life. On diesel engines with the unit operating at a pressure ratio up to 1.6 : 1 (one-hour limit rating), the maximum inlet temperatures experienced are in the neighbourhood of 500° C. to 550° C. The nozzle assembly consists of separate blades located in inner and outer rings. The inner ring is secured to the turbine inlet casing, and the outer ring is located axially but free to expand radially. The outer nozzle ring is partially slotted alternately from front and back edges to allow it to expand without distortion.

Design of Centrifugal Compressor

The centrifugal compressor comprises a single-sided impeller in an aluminium-alloy casing containing a fixed vane diffuser. Two forms of air intake are available; one of the axial type incorporating a filter-silencer for inhaling air from the immediate surroundings, and the other a single-entry volute type for use when ducting is employed. The delivery is arranged in the form of a spiral volute with a single discharge.

Air inhaled enters the impeller near its axis, as shown in Fig. 3, and while flowing out through the impeller it is given high tangential velocity. The air then enters the diffuser in a spiral path between

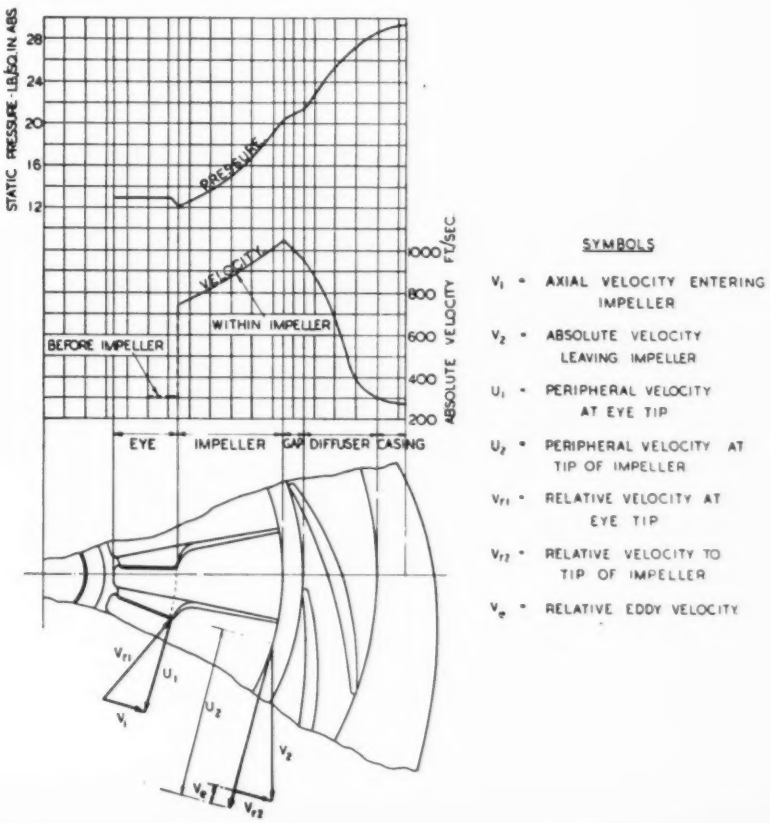


Fig. 3. Typical pressure and velocity changes through the centrifugal compressor

the fixed vanes in which the kinetic energy imparted to the air is converted into pressure with rapid reduction of velocity. The air leaving the diffuser passes through the delivery volute to the engine induction system.

The impeller has straight radial vanes which are given a degree of curvature at the entry in order that the air may enter the impeller with minimum pressure loss. The velocity diagram at impeller entry is superimposed on Fig. 3.

The tip speed for the compression ratio required has to be determined and the diameter fixed in relation to the overall dimensions of the unit and the maximum safe rotational speed for the required degree of mechanical reliability. The entry diameter for the impeller is decided in relation to the Mach No., which should be a minimum for optimum efficiency, and it is preferable for the diameter to be slightly larger than the optimum. The number of blades selected is a compromise ; a large number reduces the separation between blades but results in restriction of entry to the impeller with resultant high pressure loss. The area at the periphery of the impeller should be as small as possible provided that it does not result in too large a spiral angle. The velocity diagram and spiral angle at the periphery of the impeller resulting from the effect of the relative eddy velocity, but not from separation, is given on Fig. 3.

The tangential component leaving the impeller, for the radial blade type, is less than the peripheral velocity due to relative eddy. The angle of the air leaving the impeller is determined from the tangential velocity and the mean radial velocity. The radial distance between the impeller and diffuser entry is usually set at approximately .05 of the impeller diameter, and the outside diameter of the diffuser at 1.4 of the impeller diameter. These ratios have been established as a result of compressor research. The number of blades for the diffuser is decided as a result of previous experimental data and in relation to the spiral angle of the air entering the diffuser. The vanes approximate to an aerofoil section and the forward portion is designed to follow the spiral angle curve up to the throat formed by two adjacent blades. From this point the vane contours are designed with a certain degree of divergence from the spiral angle curve in order forcefully to divert the air from its free spiral path and so to hasten the transformation of kinetic energy into pressure. The design of blades at exit is such that the velocity of the air leaving the diffuser is reduced to within 200 to 300 f.p.s. Typical pressure and velocity changes of the air passing through the centrifugal compressor are shown in Fig. 3.

Design of Turbine

The axial-flow turbine comprises a single row of nozzle guide vanes and a rotor wheel with impulse blading. To avoid return flow of the exhaust gases to adjacent manifolds, certain of the nozzle guide vanes mate with the dividing walls in the intake casing. In designing the turbine the gas mass flow is assumed to be equal to the air mass flow of the compressor, and to determine the expansion ratio a pressure drop across the engine is assumed ; the mean turbine inlet temperature is determined from the diesel engine cycle. Since the compressor power input is equal to the turbine power output, the total head temperature drop across the turbine can be derived. Using these data the turbine blading is designed on the free vortex theory in which the angular momentum is assumed constant at all radii, and to satisfy this the swirl velocity varies inversely as the radius, and the axial velocity and work output are constant over the annulus.

Turbine blading for impulse conditions has all the available pressure energy converted into kinetic energy in the nozzles. As the blading is designed on the free vortex theory, impulse conditions exist only at the root where the pressure drop across the rotor blade is zero. Up the height of the rotor blading the pressure drop increases progressively to the tip to establish radial equilibrium.

For the basic turbine the nozzles and rotor blades are designed with constant annulus height. To obtain the actual blade angles the conditions at the root are first established. The nozzle outlet angle is chosen to be in the neighbourhood of 20° and from this, and the absolute velocity corresponding to the design expansion ratio, the axial velocity at exit is determined. Since this axial velocity is held constant, the annulus height can be established. Having determined the annulus height, the axial velocity at entry to the nozzles is

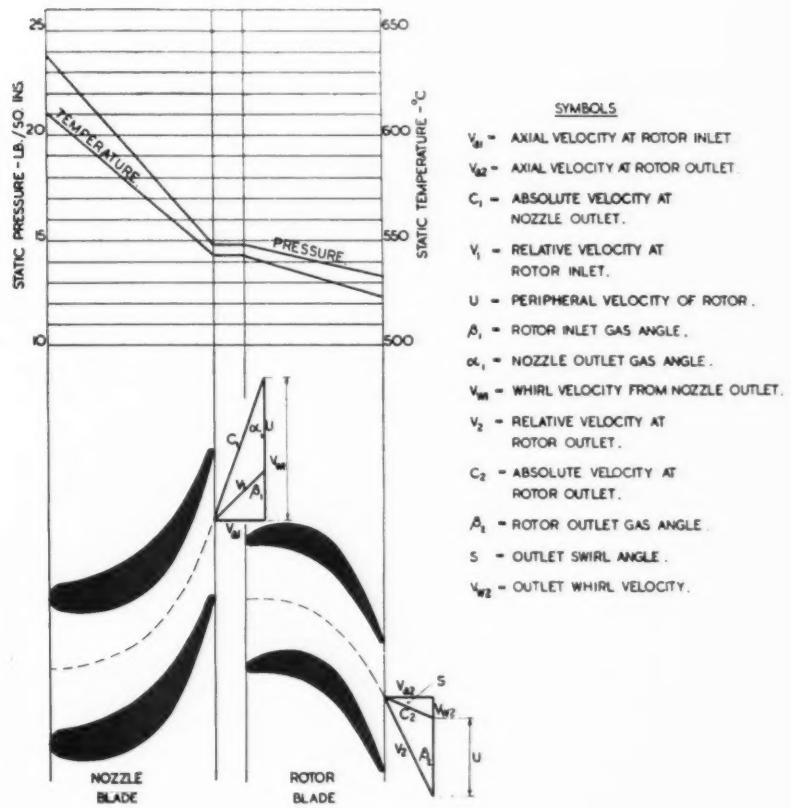


Fig. 4. Typical pressure, temperature and velocity changes through the axial-flow turbine

derived and is between 400 and 500 f.p.s. The rotor blade inlet angle is determined from the rotational speed and the nozzle exit velocities by means of the usual velocity triangles. The rotor blade outlet angle is determined from the necessary reduction in whirl component, rotational speed and the axial velocity at exit by means of velocity triangles. Swirl angles above 15° and high absolute leaving velocities are avoided in the design to minimise the pressure losses in the delivery chamber, and hence ensure maximum total head expansion ratio. Typical velocity triangles and pressure and temperature changes in the gas passing through the nozzle and rotor blading at mean blade height, are shown in Fig. 4.

Static cascade tests have established that passage shape between adjacent blades is the important design criterion and therefore, having established the gas angles, the blade sections are constructed to form convergent passages in accordance with research data. To ensure high efficiency over the operating range of the turbine some incidence on to the rotor blading is allowed for in the design conditions and therefore the blade angles differ slightly from the gas angles derived from the velocity triangles.

Matching Unit to Diesel Engine

The choice of a turbo-compressor unit for a specific engine is made by considering details of the engine type, capacity and rating ; using these data, the basic size of unit, the diffuser and the turbine blade height are selected and the manifolding arrangement recommended.

In deciding the free air capacity required, an air quantity in excess of the swept volume of the engine is allowed for scavenging purposes, this excess

being 35 per cent. for the 50 per cent. supercharge rating. To obtain this air consumption with the recommended valve timing a boost pressure of approximately 4.5 lb/sq. in. is required, and therefore a compression ratio of 1.3 : 1 is necessary from the compressor for the 50 per cent. supercharge rating. As a result of many tests on different types of engines it has been established that under these conditions good economy and relatively low cylinder exhaust temperatures are obtained.

From these derived data the basic size of turbo-compressor and diffuser are selected. The actual capacity of the turbine within this basic size of unit is determined in relation to the compressor performance characteristics and is selected to give maximum compressor efficiency for surge-free conditions over the engine operating range. This is illustrated in Fig. 2.

In recommending the manifolding for a specific type of engine, it is important to realise that the satisfactory performance obtained with low boost pressure on a turbo-charged four-stroke diesel engine is mainly due to the fact that the high energy exhaust pulses obtained at exhaust release are utilised. It is therefore essential to arrange the manifolding so that the exhaust pulses from each cylinder do not interfere with the pulses from the other cylinders and thereby reduce their effect and impair the scavenging of the cylinder. When turbo-charging four-stroke diesel engines to achieve the required excess air ratio, an increased overlap is necessary. The valve timing normally recommended for turbo-charged diesel engines is shown

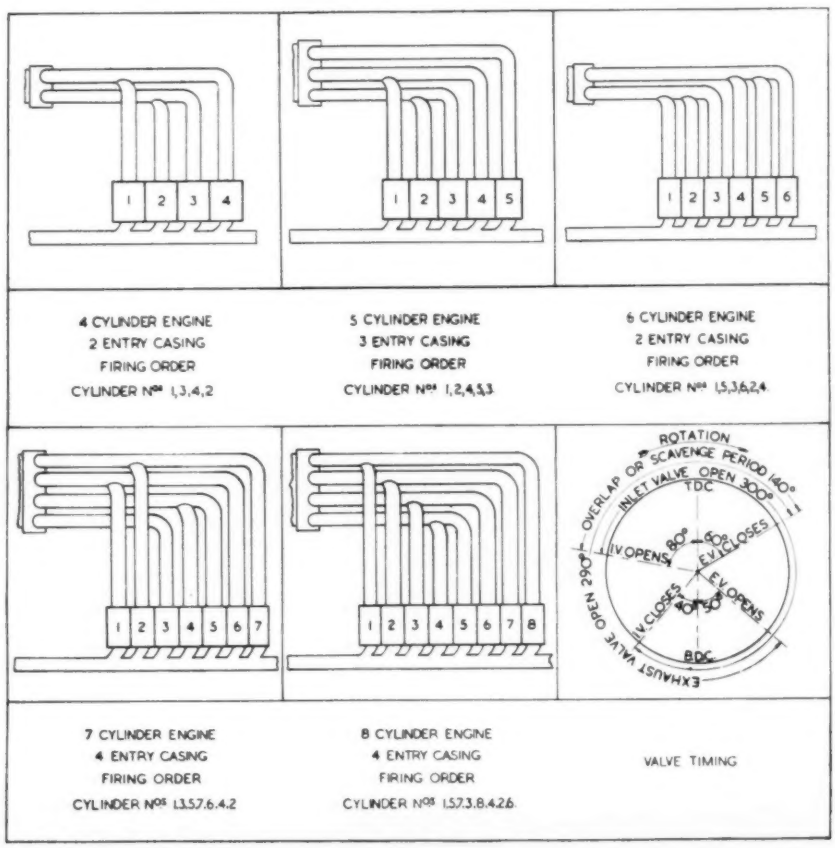


Fig. 5. Typical exhaust manifolding and valve timing for turbo-charged four-stroke diesel engines

in Fig. 5. During the overlap period the pressure in the exhaust manifold must be less than the charging pressure to obtain effective scavenging and to ensure that a minimum of residual exhaust gases is trapped in the cylinder. Ideally, one manifold per cylinder should be employed, but to reduce manifold complications and turbine annulus size it is usual to arrange that two or three cylinders discharge into one manifold and connect with one of the inlets of the turbine inlet casing which is an independent passage right up to the turbine nozzle. The maximum number of cylinders that can be discharged into one manifold is three. Where two or three cylinders discharge into one manifold the cylinders must be selected in relation to their firing order so that the points of exhaust opening are evenly spaced over two crank revolutions. In the case of three cylinders discharging into one manifold the exhaust valve of one cylinder will open approximately 40° crank angle before that of another cylinder of the group closes; this alters the pulse configuration in the manifold, but in practice it has been found that this effect can be tolerated.

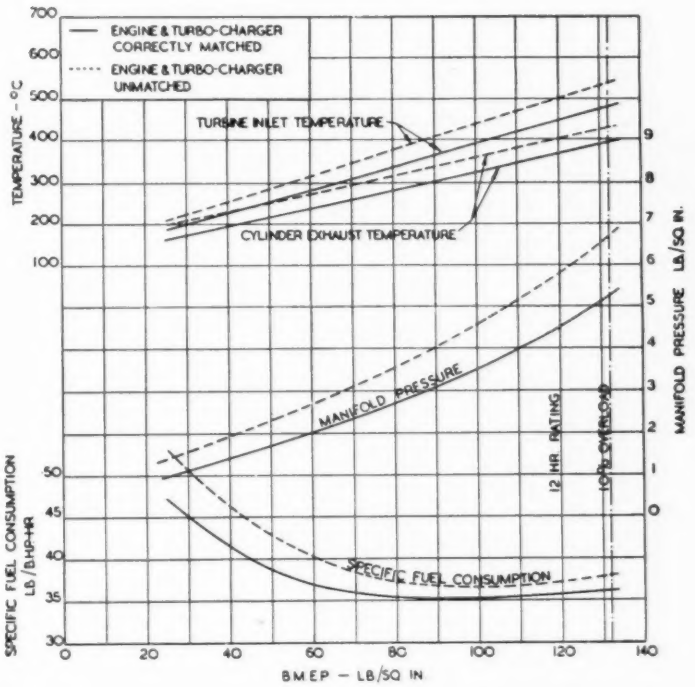


Fig. 6. Typical performance of four-stroke diesel engine with incorrectly and correctly matched turbo-compressors

For four- and six-cylinder engines two manifolds are normally recommended; for five- and nine-cylinder engines three manifolds are recommended; and for seven- and eight-cylinder engines four manifolds are recommended. An illustration of the different manifolding in relation to cylinder firing order, and the appropriate turbine inlet casing required, is shown in Fig. 5.

During initial engine trials with the recommended size of turbo-compressor and type of manifolding, test data are obtained. From analysis of these data it can be ascertained whether or not the turbo-compressor is correctly matched to the engine. Should the matching be incorrect the turbo-compressor will either be running at too low a speed and supplying too low a boost pressure, or running at too high a speed and supplying too high a boost pressure; in either case this will adversely affect the engine performance and the overall specific consumption of the engine. In view of the wide operating characteristics of the centrifugal compressor, mis-matching can generally be corrected by a change of turbine capacity. The turbo-compressors are so designed that this change of turbine capacity can be made on site and within a matter of hours. An example of the effect of mis-matching a turbo-compressor unit in relation to a diesel engine is shown in Fig. 6. In this figure curves are given for a case where the turbine is unmatched, and it can be seen that for the 100 per cent. load condition (50 per cent. supercharge) a boost pressure of 5.75 lb/sq. in. and a cylinder exhaust temperature of 400° C. are obtained, the specific fuel consumption being 0.375 lb/b.h.p.-hr.

Referring to Fig. 2, it is apparent that in this case the engine operating point is too close to surge

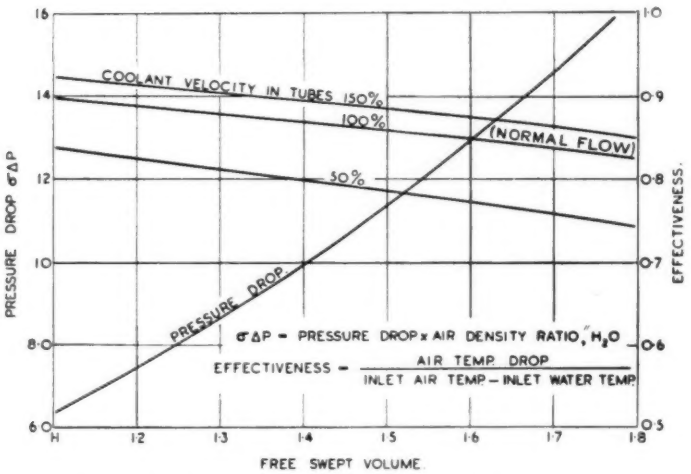


Fig. 7. Typical characteristics of water-cooled charge air-cooler

and therefore an increase in turbine capacity is necessary ; the degree of increase in capacity can be assessed in relation to the compressor characteristics. Fig. 6 shows the performance obtained after the turbo-compressor has been correctly matched to the engine, and that the boost pressure has been reduced by 1.25 lb/sq. in. to 4.5 lb/sq. in., the cylinder exhaust temperature by 35° C. to 365° C., and the specific consumption by 0.025 lb/b.h.p.-hr. to 0.350 lb/b.h.p.-hr. There is also a corresponding drop in turbine inlet temperature but this is not so important because the turbine is designed for a maximum temperature which is well in excess of the operating conditions at present experienced on diesel engines.

It will be apparent that correct matching of the turbo-compressor to the engine is extremely important, as although both the engine and turbo-compressor may be efficient individually, the overall performance obtained can be poor if they be unmatched. Once a certain type of diesel engine has been correctly matched with a turbo-compressor, unless there is a change of engine rating, individual matching of other engines of the same type should not be necessary.

Factors Affecting Supercharging of Diesel Engines

Satisfactory turbo-charging of four-stroke diesel engines depends on certain main limitations and factors, and the effect of these can be accentuated by the ambient conditions prevailing.

The limitations are engine exhaust valve temperature and maximum cylinder pressure, which limit the degree of supercharging. Supercharging can, however, be increased within the engine exhaust limitations by charge cooling, in which case the limitation tends to become mechanical loading and consequently b.m.e.p.

The factors affecting performance are variations of ambient pressure, temperature and humidity, inlet and exhaust valve timing, type of combustion chamber and the arrangement of induction and exhaust manifolding between the engine and turbo-compressor. The variations of ambient conditions are important factors where an engine is required to operate at altitude or in the tropics, as under these conditions a lower output is obtained unless the standard exhaust valve temperature of the engine is exceeded. The valve timing must be arranged to give sufficient overlap of inlet and exhaust valve opening periods to ensure satisfactory scavenging and reduction of exhaust temperature, see Fig. 5. The other two factors mentioned can have adverse effect on the performance if not given careful consideration in the initial design.

Charge-cooling, which has recently been introduced to permit increased supercharging in order to obtain higher outputs, has shown to particular advantage, since the increased outputs have been obtained without adverse effect on specific fuel consumption. For each basic type of turbo-compressor unit a water-cooled charge-cooler is available having an effectiveness of at least 70 per cent. and is designed so that it can be installed between the turbo-compressor and the induction manifolding. Fig. 7 shows the typical performance of a water-cooled charge-cooler used in conjunction with turbo-compressors.

Turbo-charged four-stroke diesel engines have world-wide application for stationary, marine and traction purposes and therefore the effect of

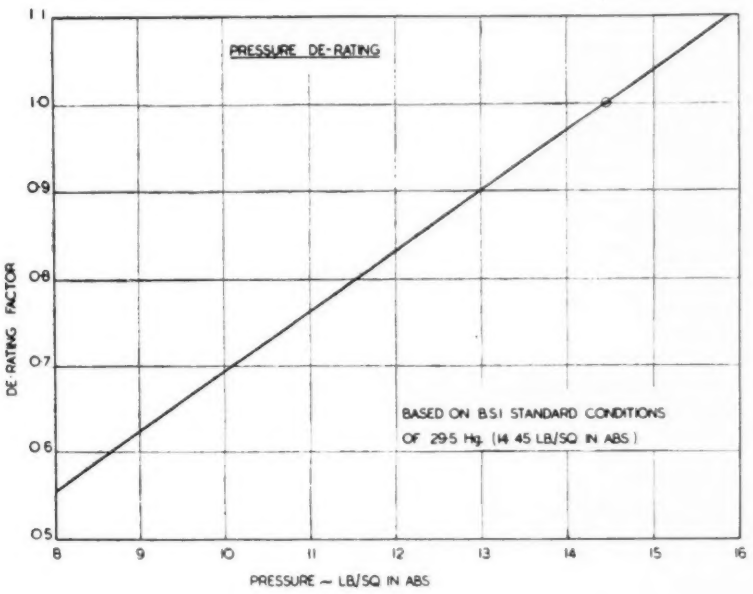


Fig. 8. Power output corrections for variation in ambient pressure

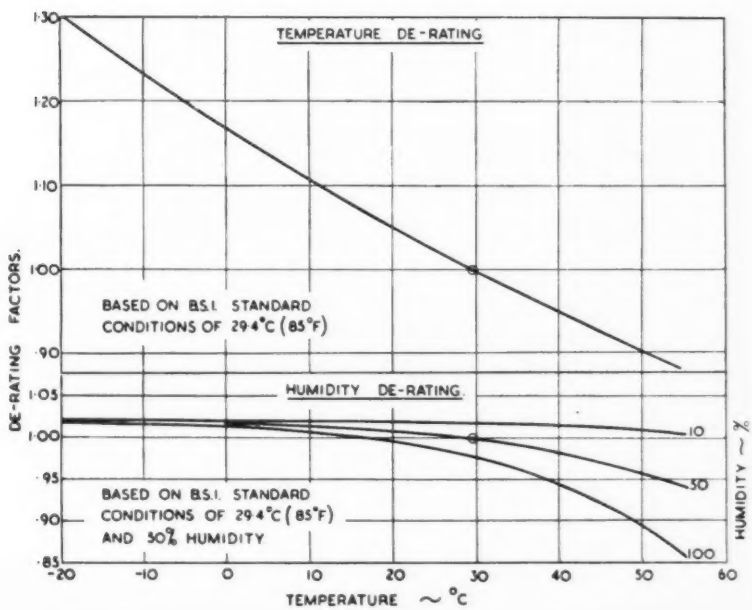


Fig. 9. Power output corrections for variations in ambient temperature and humidity

prevailing atmospheric conditions on performance must be assessed when deciding the engine rating. Variation in atmospheric pressure affects the output in direct proportion to the absolute value. Variation in temperature affects the output as an inverse function of the absolute temperature. Humidity variations affect the engine output as a direct function of the partial pressure of the water vapour. Correction curves for the effect of atmospheric conditions on engine power output are given in Figs. 8 and 9.

A turbo-charged traction engine operating under conditions of 4,000 feet altitude at a temperature of 40° C. with a humidity of 50 per cent. would have an output 18 per cent. lower than the standard B.S.I. rating if the established cylinder exhaust temperature is not to be exceeded. Therefore, for such an application, an engine must either be de-rated or allowed to operate at a cylinder exhaust temperature higher than for that of the standard B.S.I. rating. The standard B.S.I. conditions for rating purposes are 29.5 in. Hg (14.45 lb/sq. in. absolute), 29.4° C. (85° F.), and 50 per cent. humidity (15 mm. Hg vapour pressure).

Trend of Future Development

It is anticipated that the trend of future development in the four-stroke diesel engine field will be towards the introduction into service of turbo-charged engines with power outputs twice those of normally aspirated engines of the same capacity. In view of this a new series of turbo-compressors has been developed for a maximum continuous rating of 2.0 : 1 pressure ratio, and these are now available to the industry together with suitable charge-coolers.

In the two-stroke diesel engine field a number of systems of supercharging by means of turbo-compressors are at present being tried, including various versions of the parallel and series systems incorporating mechanically driven compressors with the object of obtaining satisfactory starting and low output operation. The inherent difficulty in turbo-charging two-stroke diesel engines is the fact that until a certain engine speed and power output are reached the turbo-compressor is not self-supporting. The trend of development in this field would appear to be the establishment of a supercharging system incorporating a turbo-compressor in which a reasonably simple installation is obtained and good performance over the operating range achieved with a high standard of mechanical reliability.

The Napier 'Nomad' Aero Engine

THE NAPIER 'NOMAD' aero engine, of which two views are shown in the accompanying illustrations, is a 'compounded' power unit combining a highly supercharged two-stroke compression-ignition engine with an exhaust-driven gas turbine.

This combination of piston engine and gas turbine has potentially an extremely high thermal efficiency with correspondingly low fuel consumption, and the consequent saving in the weight of fuel to be carried permits extended flight ranges or increased pay-loads. The primary purpose of the 'Nomad' is therefore to provide economical air transport over long distances.

As exhibited at the Farnborough display of the Society of British Aircraft Constructors in 1951, the 'Nomad' comprises a 12-cylinder horizontally opposed compression-ignition engine which drives one half of a counter-rotating propeller, the engine being supercharged by an axial compressor and a centrifugal compressor in series. Exhaust gas from the engine is fed into a gas turbine which drives the other half of the propeller. Two auxiliary combustion chambers are fitted in which additional fuel may be burned to provide maximum power for take-off, under which condition an auxiliary turbine is used.

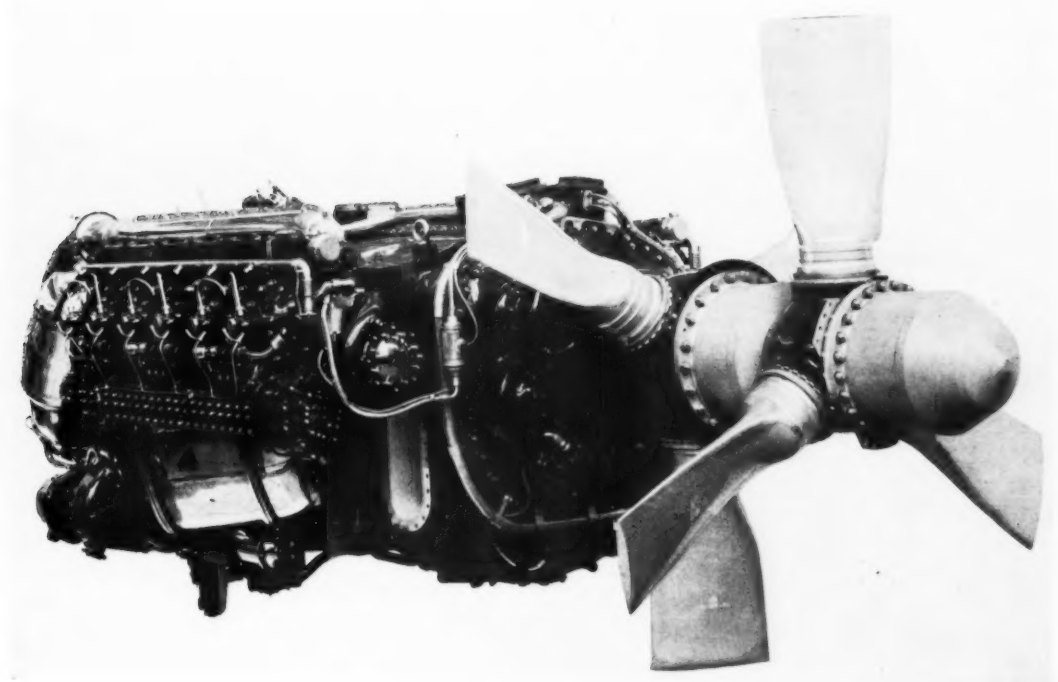


Fig. 1. *The Napier 'Nomad' engine, from propeller end*

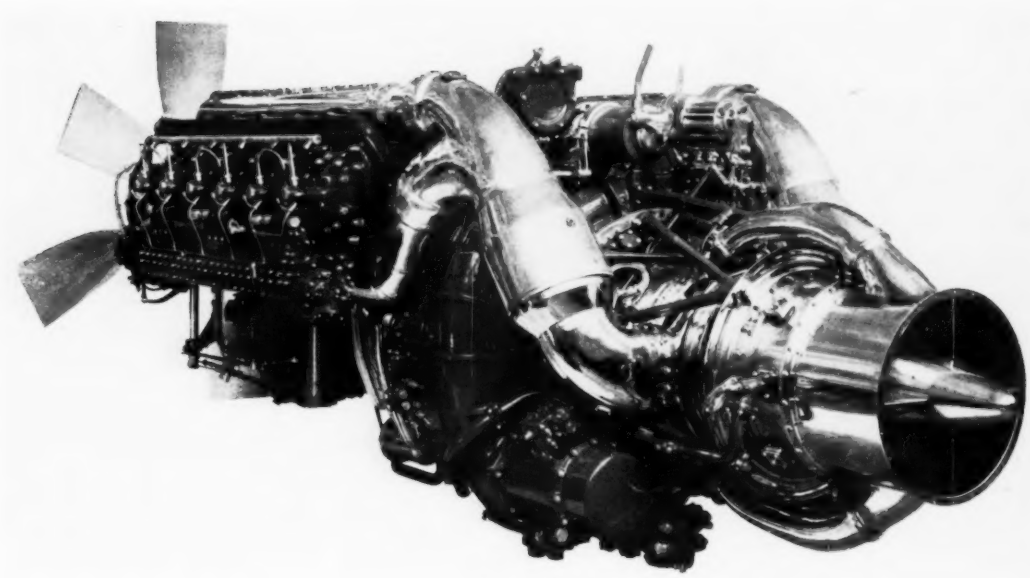


Fig. 2. The Napier 'Nomad' engine, from intake end

Leading particulars of this engine are :—

Power for take-off (sea-level, static)	3,000 s.h.p. plus 320 lb. thrust.
Fuel consumption at maximum continuous power (sea-level, static)	0.360 lb. per effective h.p. per hour.
Net dry weight ..	4,200 lb.
Overall dimensions ..	Length 126½ inches.
	Width 58¼ ,,
	Depth 49¼ ,,